Master thesis

Modélisation et dimensionnement étanchéité statique GMP hybrides et électriques/ Modeling and sizing of static GMP hybrid and electric seals

LANOS Antoine

CTU Master of Automotive Engineering

ENSTA Bretagne Architecture des véhicules

Internship completed from 03/02/2020 to 31/07/2020

Internship supervisor: LELEU Mathieu

Internship tutor : ELMESKINE Youssef

ENSTA Bretagne
2, rue François Verny
29806 Brest Cedex 09
France

CVUT
166 27 Praha 6
Dejvice-Praha 6
Tchéquie

Renault : Technocentre
1 avenue du Golf
TCR RUC 3 31
78 084 Guyancourt CEDEX
France

MASTER'S THESIS ASSIGNMENT

I. Personal and study details

Student's name: **Lanos Antoine** Personal ID number: **489811**

Faculty / Institute: **Faculty of Mechanical Engineering**

Department / Institute: **Department of Automotive, Combustion Engine and Railway Engineering**

Study program: **Master of Automotive Engineering**

Branch of study: **Advanced Powertrains**

II. Master's thesis details

Master's thesis title in English:

Modelisation and dimensioning of sealing of electrical and hybrid powertrain

Master's thesis title in Czech:

Modelování a dimenzování těsnicích prvků v elektrických a hybridních převodných ústrojích

Guidelines:

Make a literature survey of used sealing types, and its usage in automotive transmissions. With the new type of powertrains, which appear with the electrification and hybridisation of vehicles, occur new demands also on sealings. Take into account the new functional demands and limits and make an overview on its impact on sealing. Make a survey of existing techniques used by Renault. Conceive the methodology and procedures for dimensioning and right choice of the sealing type. Show at least 1 example of defective case, propose the correction. Document the proposition with detailed drawings of concerned parts.

Bibliography / sources:

Name and workplace of master's thesis supervisor:

doc. Dr. Ing. Gabriela Achtenová

Name and workplace of second master's thesis supervisor or consultant:

Christophe Chalu, Renault

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Assignment valid until: ____________

doc. Dr. Ing. Gabriela Achtenová Supervisor's signature	doc. Ing. Oldřich Vítek, Ph.D. Head of department's signature	prof. Ing. Michael Valášek, DrSc. Dean's signature

III. Assignment receipt

The student acknowledges that the master's thesis is an individual work. The student must produce his thesis without the assistance of others, with the exception of provided consultations. Within the master's thesis, the author must state the names of consultants and include a list of references.

05/08/2020 — Date of assignment receipt

LANOS Antoine — Student's signature

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Finally, my special thank goes to Youssef Elmeskine for guiding me as much as necessary, for taking the time to share his knowledge and experience with me, and for making this internship motivating and enriching despite the difficult context.

List of abbreviations

AMI: Asia, Middle east, India
CIPG: Cured In Place Gasket
EMC: Electromagnetic Compatibility
EMF: Electromagnetic Field
EMI: Electromagnetic Interferences
EPDM: ethylene propylene diene monomer
E-PWT: Electric Powertrain
EV: Electric Vehicles
FIPG: Formed In Place Gasket
ICE: Internal Combustion Engine
IRHD: International Rubber Hardness Degrees
LEM: Liquid Elastomer Molding
PEC: Power Electric Controller
PWT: Powertain
RSM: Renault Samsung Motors
TH-PWT: Thermic Powertrain

Notations list

e: Part of gasket's total length (m)
E: Young Modulus (Pa)
f: Friction coefficient
F: Force (N)
H: Groove's height (m)
H_g: Gasket initial height (m)
I: Moment of Inertia (m^4)
L: Groove's width (m)
L_g: Initial gasket's width (m)
N: Normal force (N)
P: Pressure (Pa)
S: Surface (m^2)
T: Tangential force (N)
x: Parting line opening (m)
ε: Compression ratio

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Introduction

Renault Group [1]

A carmaker since 1898, the Renault group is a leading manufacturer of an international multi-brand group that brings together the brands Renault, Dacia and RSM, Alpine and LADA. Present in 134 countries, the Group sold nearly 3.8 million vehicles in 2017, a record year, making it the number one French automotive group in the world. In 2018, the Renault group aims to grow driven by the development of its international activities and by its renewed range.

FIVE COMPLEMENTARY BRANDS:

- Renault is sold in 134 countries through more than 12,000 sales outlets. In its 120-year history, Renault has forged its identity on innovation accessible to the greatest number.
- Dacia is sold in 44 countries, in Europe and in a number of countries in the AMI region (Africa, Middle East, India). Since 2004, it has attracted more than five million customers, offering a range of simple, reliable vehicles at the best possible price.
- RSM is present in South Korea. The brand sells 7 sedans (including one electric model) and SUVs (sport utility vehicles).
- Alpine returns in 2017 with the new Alpine A110.
- LADA has been a Renault Group brand since January 2017 and is the historical leader of the Russian market.

The evolution of the automotive sector is perpetual. The most striking current illustration is the search for performance which, having long been the main concern, is now undergoing a real turnaround. Recent economic and environmental concerns are nowadays leading to new orientations, such as the reduction of pollutant emissions or the reduction of sales prices.

Renault accomplished that becoming the sales leader and pioneer of 100% electric mobility in Europe. The Renault group is constantly improving its electric range to meet its customers' expectations, particularly in terms of autonomy. To do so, the Group draws on its nine years of expertise and leadership in the following areas the electric vehicle and continues to renew its existing models: Renault ZOE, Renault Kangoo Z.E., as well as the Renault Samsung Motors SM3 Z.E. in Korea. (Figure 1: Renault ZE range)

Figure 1: Renault ZE range

My internship

During my internship, I was welcomed at the Technocentre in Guyancourt. The Technocentre is the leading automotive research and development centre in Europe. Since 1998, it has brought together all the players involved in the design and development of Alliance and Renault group vehicles. It is at the Technocentre that the ranges of the Renault, Dacia and Renault Samsung Motors brands are defined.

Inside the Technocentre site, I joined the department "Métiers bas moteur" made up of specialists in a specific field and whose role is to develop the technologies used and define sizing rules. The team is composed by 12 persons that are specialist in a specific field (hot and cold sealing, lubrication, assembly, batteries, static testing).

In this team, I worked with Youssef Elmeskine on static cold sealing.

Problematic

By joining the team to do this internship, the main goal was to work on static cold sealings for E-PWT. With the electrification and hybridization of powertrains, new functional demands appear on sealing.

At first, a literature survey of used sealing technologies and their application in automotive powertrains will be done to learn and understand better all static cold sealing technologies. This part will explain what cold static sealing is and detail the 3 mains sealing technologies/principles used for these sealing needs.

Then, we will take into account the new functional demands and limits and make an overview on its impact on sealing.

This will be completed by a benchmarking part that will help to understand more precisely sealing technologies and to define sizing parameters. This will be a way to identify which sealing types are used for which applications and to observe sealing strategies from competitors. Finally, we will try to identify some differences between E-PWT and TH-PWT sealings.

Then, a defective case will be studied. First, possible root causes for the problem will be investigated and then a correction will be proposed starting by analysing the functional demand and then by sizing a design that would respect these demands.

We will conclude by a review of what has been done during the internship, personal and technic benefits.

1 Bibliography

1.1 GMP Sealing

1.1.1 Generalities

Definition:

Sealing: Sealing represents the confinement of fluid, dust, or filings in an enclosure. An enclosure is said to be sealed if no particles of fluids or solids present can enter or leave it. Otherwise, there is a leak. A perfect seal is impossible to obtain, due to a certain permeability of the materials used, possible defects or the stresses to which the parts to be sealed are subjected. The materials and parts used to achieve a seal are generally called gaskets.

ICE: The engine, a combination of the combustion system and its accessories (power supply, cooling, ignition, etc.), is the main component of a vehicle, but also probably the most complex. It is made up of many parts and works partly thanks to the combination of several fluids, including air, fuel, exhaust gases and various oils. For this reason, it is necessary to make several seals on the engine using very different technologies depending on the constraints encountered for each application. The complexity and the multitude of sealing to design can be observed below as every part need to be sealed in order not to leak. (Figure 2: ICE)

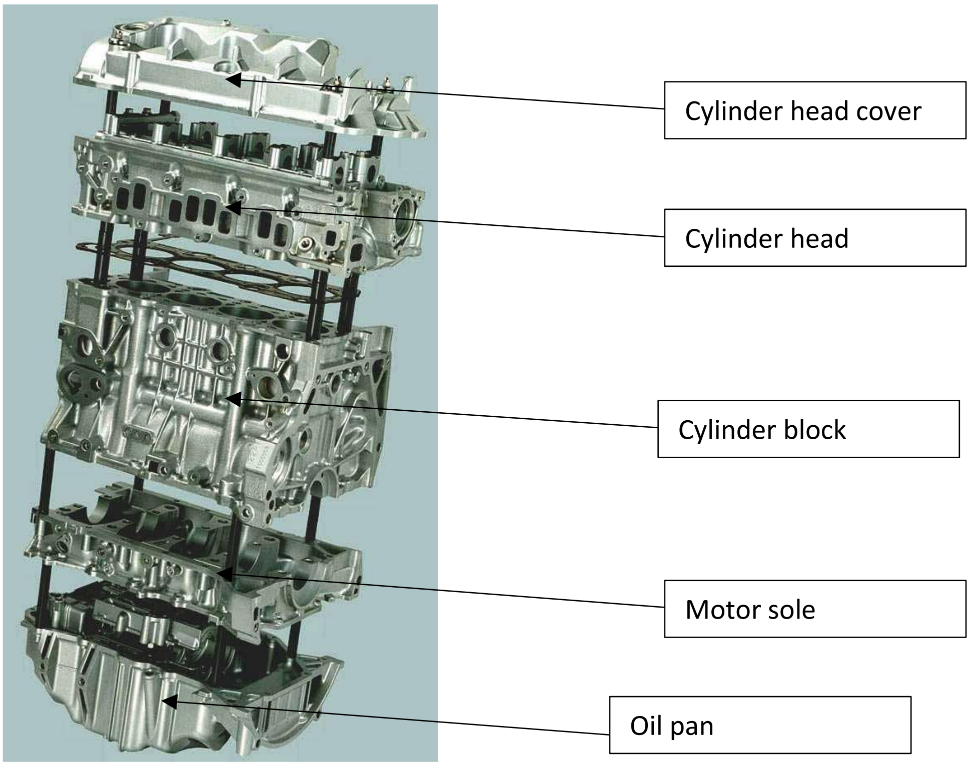


Figure 2: ICE parts

Electric powertrain: the difference between a thermic and an electric powertrain is the use of an electric motor instead of a thermic motor. That implies the use of new cooling and lubrication technologies, it is synonym of new constraints and as a result new sealing problematic. As shown below, electric powertrain structure differs from thermic powertrain. (Figure 3: Electric powertrain) Cooling is needed to cool down the power electronic unit and depending on the technology used lubrication is used in the rotor/stator part. Here, sealings for intake and exhaust gases are not needed as there is no combustion. A new type of need on these powertrains is often the complete sealing of the battery system.

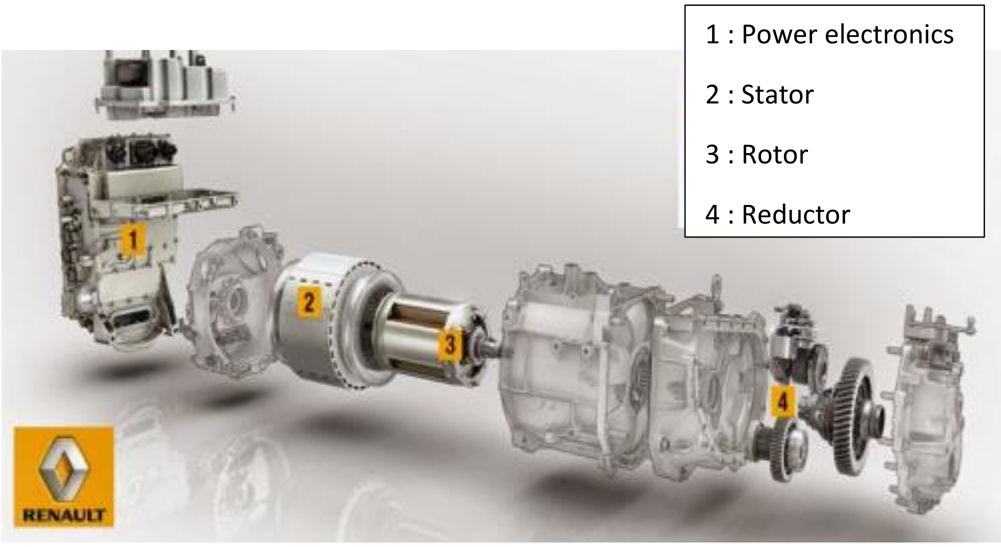


Figure 3: Electric powertrain

1.1.2 Cold/ hot sealings, Dynamic/ static sealings

PWT Sealing

To seal a PWT different types of gaskets can be used. Whatever the gasket, it must have four properties:

- elastic, to follow dimensional variations between the surfaces to be sealed under the action of the various stresses (static and dynamic) and ensure in always with minimum contact effort.

- plastic, in order to adapt as well as possible to the surface conditions, and in particular those related to roughness and flatness.

- impermeable to the fluid to be sealed.

- compatible with the process fluids (machining cutting oil and detergent) and with the fluid to be sealed under all operating conditions.

Seals can be classified into two main families: static sealing and dynamic sealing

1. Dynamic/static sealings:

First, sealings are divided into two main families: static and dynamic sealings. Dynamic seals are used when sealing between parts that are moving relative to each other (the most common example being a shaft rotating in its housing, for which a lip ring is usually used). On the contrary, sealing is considered static when it is carried out between parts which are stationary in relation to each other.

There are different types of static sealings:

- Metallic and semi-metallic gaskets

These can be, flat gasket, beaded gasket and O-ring.(Figure 4, Figure 6, Figure 5)

Figure 5: flat metal gasket

Figure 6: beaded metal gasket

Figure 4: O-ring

- Non-metallic gaskets

These can be fibrous or non-fibrous gaskets.

Non asbestos gaskets: these gaskets can be rubber or cork gaskets. (Figure 7)

Figure 7: non asbestos fiber gasket

Fibrous gaskets: fibers are like a spine for the gasket, they are the structure of the gasket. Binders can be added in fibers to bind every component. Those binders can be silicones, rubbers of elastomers. Finally, fillers can be added to fill in the gaps.

2. Hot/cold sealings:

Seals can also be distinguished according to the temperatures the seals are exposed to. On the one hand, there are the so-called hot gaskets, with the famous example of the cylinder head gasket or the exhaust manifold gasket, which are in contact with gases whose temperature can go up to 900°C. On the other hand, there are the so-called cold seals, such as the oil sump or timing case seal, which are in contact with oil rarely exceeding 200°C.

The interest here will stay on static cold sealings. Sealings technologies associated with static/cold sealing will be detailed bellow in the next part.

1.2 Static/Cold sealing strategies

Three different types of static/cold sealings technologies are usually used to seal motor parts. Silicon rubber gaskets, flat metallic gaskets, and elastomer gaskets. In order to choose between these three different types of gasket, there are some selection criteria. For example, operating temperature, material compatibility, price, post-sold needs, and interfaces geometry. The presentation of each technology will help to justify the use of one type of gasket for a specific problematic.

1.2.1 Flat metallic Gasket [2]

Flat metallic gaskets are often used for hot sealing application such as cylinder-head gasket. But sometimes, it can be used for cold sealing application. The most used gaskets in this case are called elastomer-coated beaded metallic gasket.

Elastomer-coated beaded metallic gasket: (Figure 8: Elastomer-coated beaded metallic gasket)

This type of seal consists of a metal sheet, into which a bead is pressed. The sheet is coated with a layer of elastomer of a few microns.

In order to ensure the sealing, a contact pressure is needed. That pressure is provided by the bead and is the most important characteristic of the gasket. Indeed, when the bead is crushed, pressure lines are generated and ensure the macro-sealing (Figure 9).The elastomer is used in order to provide a micro-sealing by filling surface defects.

Figure 8: Elastomer-coated beaded metallic gasket

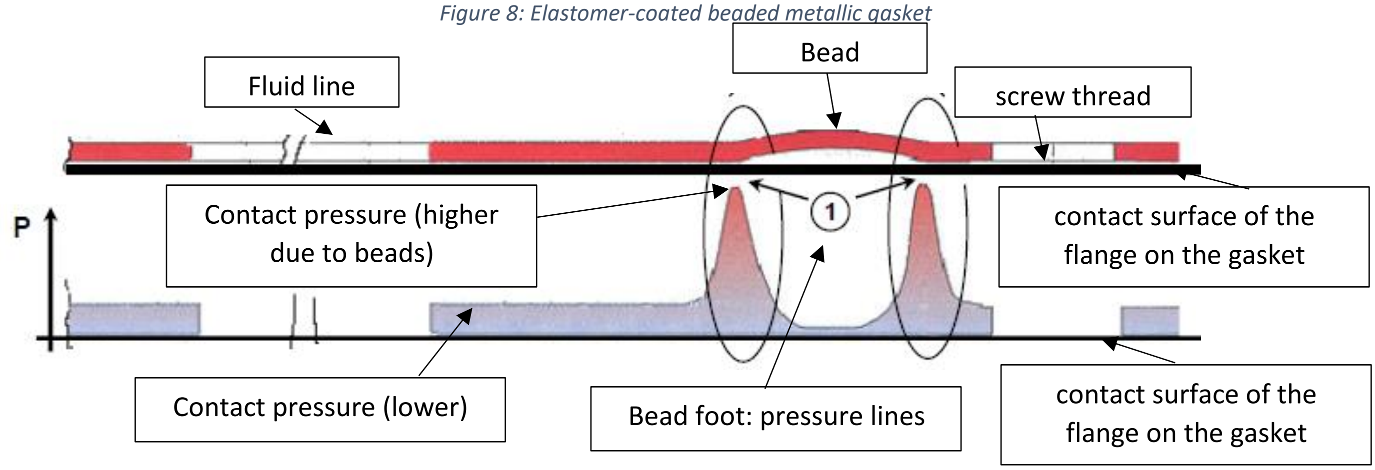


Figure 9:Theoretical pressure fields between corrugated plate and interface

With this technology, in order to ensure the sealing, a minimum contact pressure and a maximum pressure difference are required. To respect this condition, the stiffness of the gasket can be adjusted. Indeed, the stiffness variation produces a pressure difference (difference between the max pressure (pressure lines) and the minimum pressure) and a minimum contact pressure variation.

For example, if the minimum contact pressure is too low, the pressure difference has to be decreased to increase the minimum contact pressure. To do that, a decrease of the stiffness of the gasket is needed.

1.2.1.1 Parameters and materials influences

- <u>Stiffness variation</u>

There are several ways to change the gasket stiffness:

A first possibility is to change the bead stiffness.

a) Bead type (Figure 10,Figure 11)

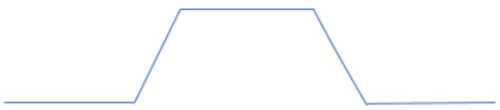


Figure 10: Bead

Figure 11: half bead

Full beads are, for the same thickness, profile, and material, stiffer than half beads.

b) Bead profile

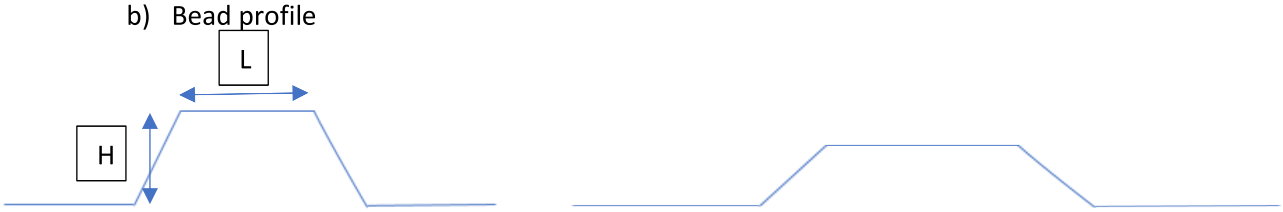


Figure 12: bead profiles differences

The pressure difference can also be increased or decreased by changing the bead profile. Indeed, when H increase the pressure difference increase and on the contrary when L increase the pressure difference decrease. (Figure 12: bead profiles differences)

c) Gasket thickness

In order to adjust the gasket stiffness, the thickness can be modified. An increase of the thickness will increase the gasket stiffness.

d) Gasket material

Finally, the material can be changed to obtain different properties and stiffness for the same geometry.

- <u>Gasket coating</u>

MOS2 coating can be added in order to limit the fretting corrosion.

LEM (liquid elastomer molding) (Figure 13)

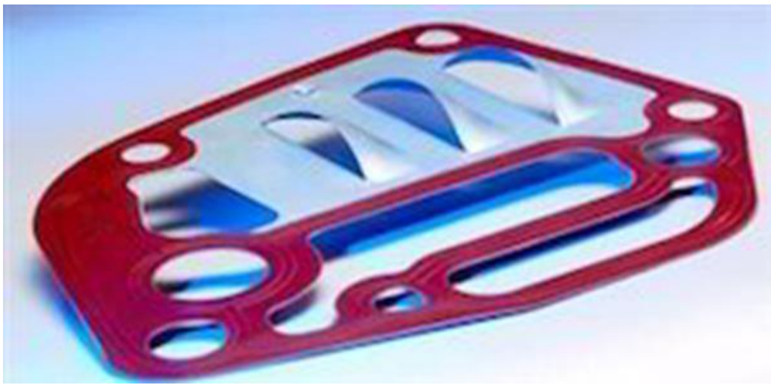


Figure 13:LEM gasket

LEM gasket are composed of a metal frame with an elastomer over moulding. The elastomeric layer is composed of beads which generate the pressure lines, it is the same principle as the flat metallic gasket.

All these geometries and properties variations are a way to respect the contact pressure criteria.

1.2.1.2 Advantages and disadvantages

Flat metallic gasket:

Flat metallic gasket is not the first choice for static sealing. Indeed, it is more expensive than elastomer gasket and silicon rubber gasket. As the reduction of the production costs is a priority, this type of gasket is not often used.

But in some cases, and for some specific reasons, flat metallic gaskets are used for cold static sealing applications:

- In case of dangers related to silicon particles detached, flat metallic gasket is recommended as it does not emit particles.
- If there are after-sales needs, silicon rubber gasket is not recommended as it is difficult to unseal and seal again. Flat metallic gasket is in this case more practical as it is easier to disassemble.
- Finally, flat metallic gasket requires a smaller minimal sealing flange dimensions compared to elastomer gaskets and silicon rubber gaskets. This can be an advantage to reduce sealing flange size and total weight.

Concerning LEM, it is more expensive, but the base elastomer coating provides a good overall micro-seal with the silicone beads ensuring optimum sealing in the critical areas. It is often used for the lower housing part, water pumps or intake manifold.

1.2.2 Elastomer Gasket [3]

1.2.2.1 Elastomer definition:

An elastomer is a polymer with viscoelasticity, and has very weak intermolecular forces, generally low young’s modulus and high failure strain compared with other materials. Each of the monomers which link to form the polymer is usually a compound of several elements among carbon, hydrogen, oxygen, and silicon.

1.2.2.2 Elastomer properties and behaviour [4]

Elastomer are materials that are stiff besides a precise temperature (Figure 14). They are used close to the absolute melting temperature of their secondary connections (Van der walls connections). Polymers have some characteristics of materials close to fusion: they flow and the elastic deformation that appears under stress, increases over time.

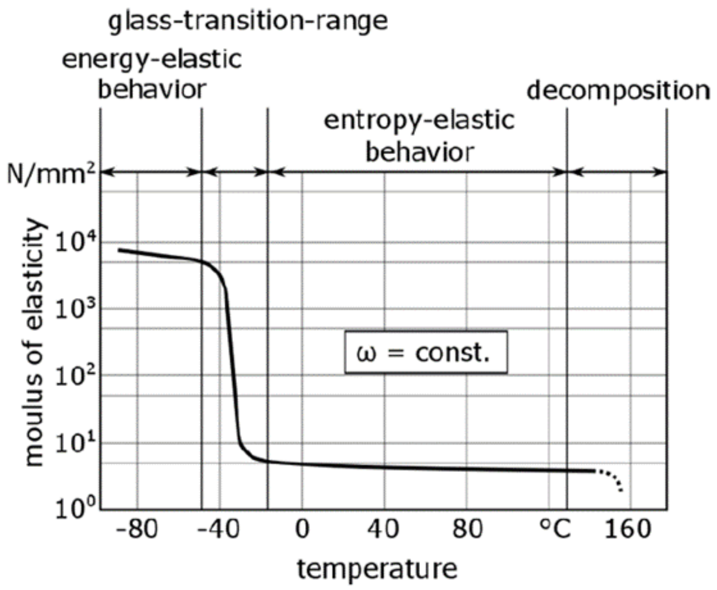


Figure 14: elastomer behavior with temperature

What is very interesting with elastomer is their elasticity. The elasticity of a rubber comes from the vulcanization process. Vulcanizing consists of "curing" an elastomer to create cross-linking. This cross-linking curing creates bridges between the chains of an elastomer. It is these bridges that give the rubber its elastic properties. (Figure 15) Cross-linking created by curing allow the material to return to its initial shape.

Initial phase	Stretching phase	Final phase

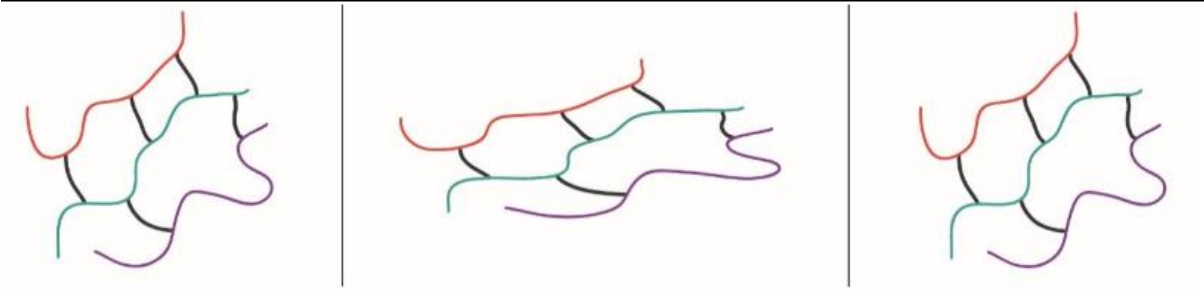


Figure 15: curing and elasticity

The behaviour of those elastomer gaskets depends on several factors:

- Light, temperature and ozone
- Fillers to reinforce the gasket and to bind all the components
- Curing density:
  - It increases the stiffness and the elasticity module of the elastomer
  - It decreases the tensile strength
- Plasticizer addition to increase high temperature performances

1.2.2.3 Different stiffness [5]

Hardness is a parameter used to define the resistance of a material to penetration by a harder body. Hardness is measured using a hardness tester which can be expressed in different scales: Shore A, Shore D and IRHD. The unit of measurement, Shore extends over a range from 0(soft) to 100(hard) (Figure 16).

Figure 16: stiffness tests [source]

In order to choose correctly the stiffness, we have to take into account:

- the maximum pressure of the fluid to be sealed
- the type of use or assembly
- adjustment tolerances

1.2.2.4 Functional criteria

- Sufficient covering surface:

The covering surface: this consists of checking that there is a sufficient covering surface between the sealing flanges of the facing pieces to ensure compression of the joint.

- Seal containment rate <1

The containment is a ratio which must always be less than 1, it is calculated taking into account the tolerances on the groove and the seal as follows:

$$\text{Containment} = \frac{S_{maxi\ joint}}{S_{mini\ gorge}} < 1$$

- Compression ratio [10-30%]

The compression ratio is a ratio between the free height of the gasket and the height of the seal when compressed in its groove. This parameter is expressed in percentage and directly conditions the quality of the seal achieved.

A check is carried out, taking into account the tolerance intervals on the groove and the seal. The compression ratio must be between 10% and 30%.

The compression ratio is calculated as follows:

$$Compression\ ratio = \frac{H_i - H_f}{H_i} \times 100$$

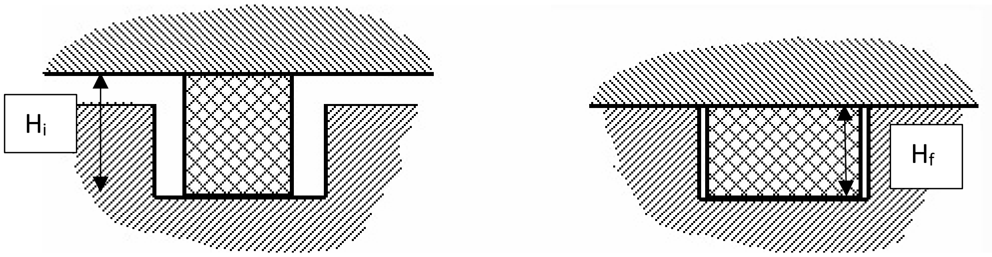


Figure 17: compression ratio

1.2.2.5 Sealing principle

The purpose of the elastomer seal is to ensure a tight seal between two parts. The sealing is achieved thanks to the deformation and reaction properties of the elastomer during the tightening of the parts assembled. The assembly is rigid when there is metal-to-metal contact between the interfaces or decoupled if one of the interfaces is in contact with the gasket, without metallic contact with the other interface. A distinction is made between soft elastomer seals mounted in a groove and elastomer seals with metal insert, mounted on flat interfaces.

There are therefore two main "families" of elastomer seals:

- Elastomer gasket in a groove

There are 3 types of elastomer gasket in a groove (Figure 18; Figure 19; Figure 20):

Figure 20: elastomer O-ring

Figure 19: Extruded/stranded

Figure 18: Moulded

In this case, all or part of the seal is placed in a groove which is made on one of the two interfaces to be sealed.

- Gaskets with metal insert (Figure 21)

Those are used on flat surfaces and have a metal frame that stiffens the gasket and the assembly.

Figure 21: Elastomer gasket with metal insert

1.2.2.6 Advantages and disadvantages:

Advantages:

- Adaptability to interfaces: The elastomer seal is a seal that adapt perfectly to the profile of the interfaces to be sealed and therefore is more able to make up for possible non-conformities and surface conditions (roughness, flatness, waviness). The operating range (surface finish tolerance interval) of a gasket in a groove is larger than that of a silicone gasket.
- Assembly feasibility and after-sale: Elastomer gasket are relatively easy to assemble and disassemble if needed.

Disadvantages:

- Cost: the product cost of an elastomer gasket is higher than cost of a silicone gasket.

- Seal behaviour: For groove-moulded seals, the principle of seal deformation is not completely mastered. The only way to approach it is by finite element calculation, which is currently limited by the lack of knowledge of elastomer behaviour curves.

1.2.3 Silicon rubber Gasket [6]

1.2.3.1 Definition: Formed In Place Gasket (FIPG)

The FIPG consists of the application of a silicone bead on the parting line of one of the two parts to be sealed. When the two parts are joined together, the product not yet dry, spread over the width of the parting line and then cured. This creates a film of product between the two rooms. Once cross-linked, the cohesion of the silicone ensures the seal.

Two types of silicon can be distinguished, non-adherent silicon and adherent silicon: (Figure 22: adherent vs non-adherent silicon)

- Non-adherent silicones:

These are curing products of the acetic or acetone type. During engine operation (temperature rise), these products develop a moderate adhesion which allows the system to be sealed, while ensuring relatively easy disassembly if necessary. Their mode of rupture is said to be "non-cohesive".

- Adherent silicones

These are curing products of the alkoxy or oxime type. These products develop a strong adhesion on a large majority of substrates: metals of all types. This characteristic allows a good resistance of the material in the case of mechanical stresses (tensile and/or shear). These products are less suitable for applications requiring frequent disassembly. The use of stops that promote disassembly by leverage effect is essential for disassembly if necessary. Their mode of rupture is said to be "cohesive" because it results from a rupture within the parting line.

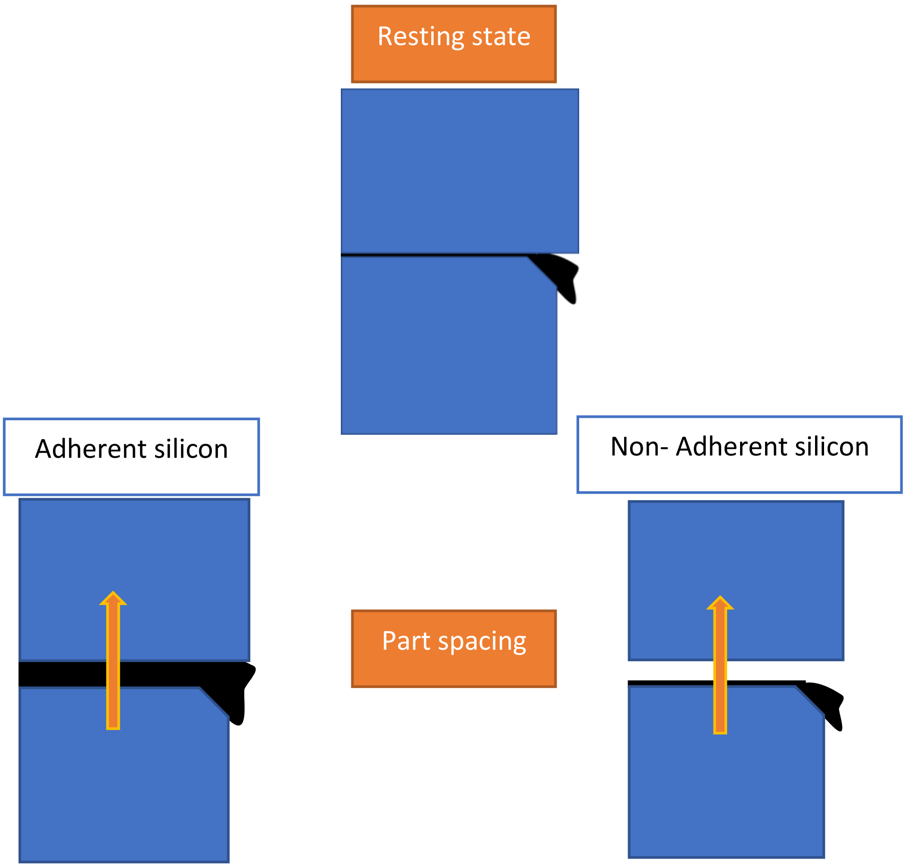


Figure 22: adherent vs non-adherent silicon

But what is the curing (cross-linking) and what properties does it give to the silicon?

1.2.3.2 Curing/cross-linking

Several types of vulcanization exist, hot-crosslinking and cold-crosslinking silicones. Hot cross-linking silicones reticulate between 110 and 160°C with the addition of peroxide. Their cross-linking is very fast.

Among the cold-crosslinking silicones (i.e. at room temperature), a distinction is made between mono- and bi-component silicones. The crosslinking of a two-component is initiated by mixing the silicone with a crosslinking agent previously stored separately. This type of cross-linking usually allows for rapid and uniform cross-linking.

The cross-linking of a mono-component is triggered by contact of the silicone with air humidity. This curing process takes more time, extern layers reticulate first as they are in contact with the air. The inside has to wait for the air humidity to propagate. This can take some time if the cord is thick. However, the use of mono-component silicones has undeniable advantages in the assembly process (a single product, does not cross-link as long as it is not exposed to the air, ...).

The cross-linking of silicones gives off various compounds, such as:

- acetic acid for acetic silicones
- acetone for acetone silicones
- alcohol for alkoxy silicones
- ketoxime for oxime silicones

The properties of cross-linked silicones depend on part of the family to which they belong, but there are also wide disparities within these families. The time required for vulcanization by example is one of these properties, as shown in the following example. (Figure 23)

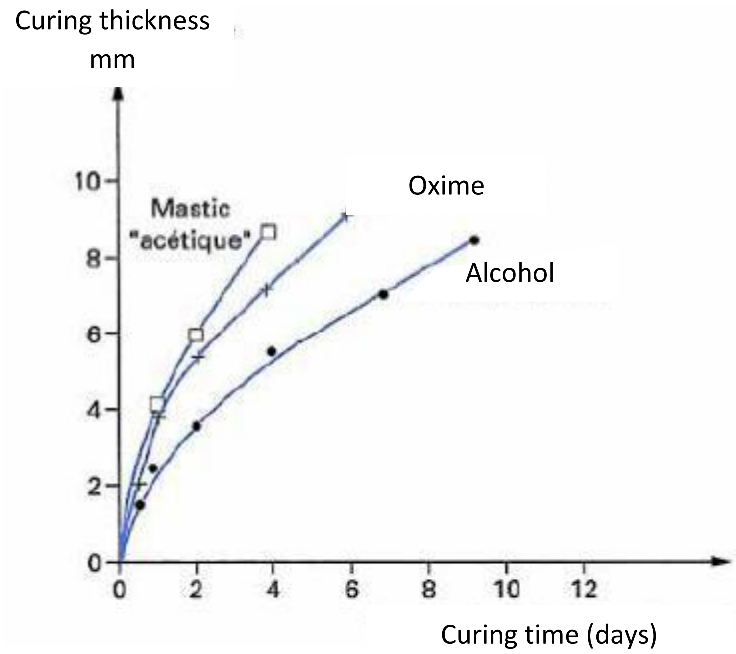
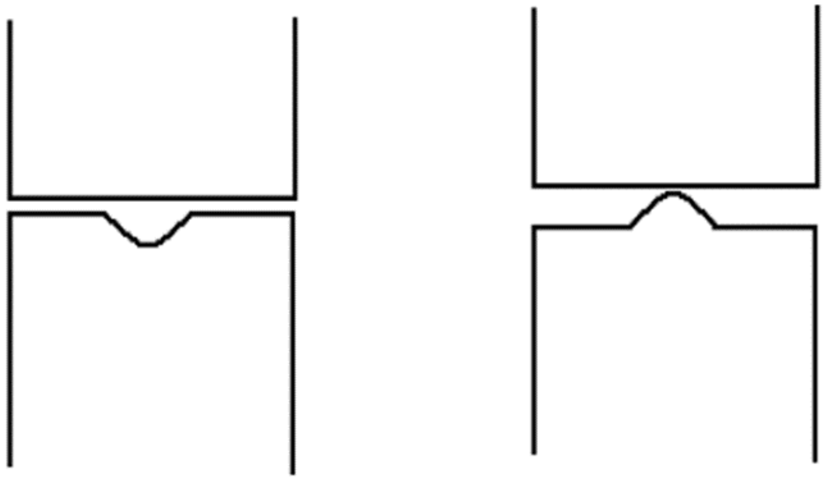


Figure 23: Curing time for different families

Following this vulcanization step, the silicone elastomer is no longer in a fluid form. During the passage on the assembly line, it is necessary to assemble the parts shortly after removal. Cross-linking starts as soon as the silicone is in contact with the air, so it is necessary to assemble before a layer of cross-linked silicone (called the first skin) is formed. Some products have a long curing time for greater flexibility in the process: it allows the 2 parts to be assembled relatively long after the bead has been deposited (around 20 minutes).

1.2.3.3 Parting line design

Some parameters and geometry are very important to increase the seal security. The parting line is the surface on which the silicon is applied. It is composed of the sealing flange, the chamfer and sometimes a boss or a groove. Minimal required sealing flange dimensions are not the same if there is a groove or a boss. (Figure 24) The boss is often a way to rigidify a part. On the other side, a groove is a way to get another elastic reserve. These two geometries can be associated with a chamfer.



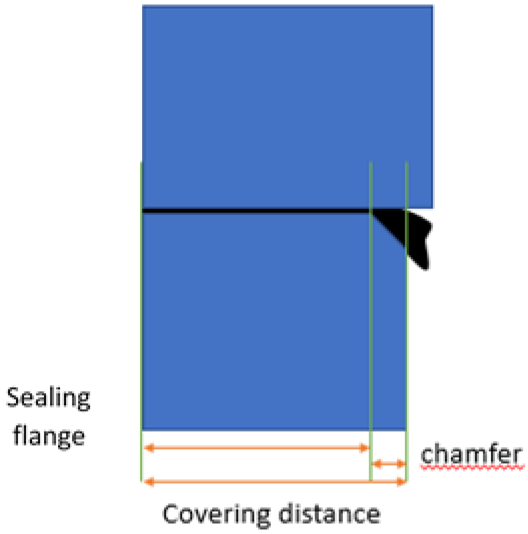


Figure 24: Parting line definition

This parting line definition will be explained more precisely in the benchmarking part.

1.2.3.4 T-joint and its risks

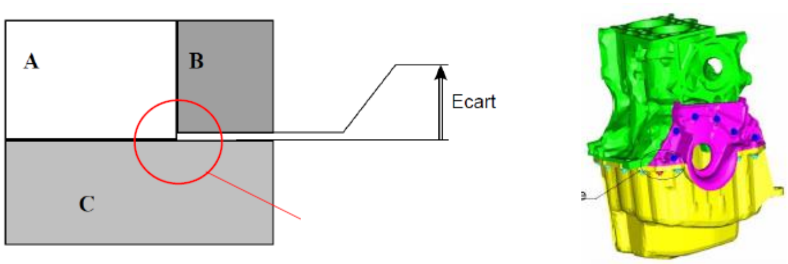


Figure 26: three way joint definition

The T-joint presents a significant risk of leakage. Indeed, when the three parts are assembled, the stacking tolerances (both on the parts themselves and on the assembly that has been made) creates a clearance, sometimes important.

Furthermore, in many cases the three parts are not made of the same material. Under the effect of heat, they do not expand in the same way. This is called differential expansion. This phenomenon creates stresses and deformations that can be added to the existing clearances. For example, aluminium expands twice as much as cast iron at the same temperature.

The FIPG must be able of compensating both the clearance due to the stacking of tolerances and the opening due to the differential expansion of the parts.

Those T-joint are often located at the junction between the cylinder block, the oil pan, hand the seed housing. (Figure 26)

In the event that there would be two different materials to seal the T-joint, these must be compatibles.

1.2.3.5 Advantages and disadvantages

Technical advantages:

- high-performance product: the products used have a good resistance to ageing, they react very little with oils and coolants, their mechanical behaviour is indifferent to temperature variations (-70 to 250°C).
- automation possible: the use of a dispensing robot allows a perfect control of the bead geometry, a very fast dispensing, and a good reproducibility.
- versatility of the dispensing means: the same equipment (dispensing robot) can produce several different profiles, without having to design a particular seal adapted to each part.

Economic advantages:

- cheaper raw material: the cost of the seal is divided by 5 to 10 compared to a moulded seal, excluding depreciation of the dispensing equipment.
- standardization: no stock of gaskets with multiple references.
- no handling (manual or robotic) of moulded joints.

Disadvantages:

- Cannot be disassembled: it is impossible to disassemble the assembly without damaging the joint, so solutions must be provided for after-sales.

1.2.4 Conclusion

The three principal types of sealing used for PWT sealing are described in this part. They all have their advantages and disadvantages and that is why some technologies are used for specific applications. The next part will present which type of gasket is used for which application.

2 E-PWT/TH-PWT sealing constraints differences

The interest will be place in the difference between E-PWT and TH-PWT from a sealing point of view. The difference between E-PWT and TH-PWT comes from the way each powertrain works. These powertrains use different technologies with different parts. Indeed, the difference of energy used implies the use of different components with different solicitations.

2.1 Thermic constraints differences

For thermic powertrains, the combustion is source of high thermic constraints for some sealing. Indeed, the cylinder head gasket and all the sealings for exhaust gazes must resist very high temperatures (900 °C). For electric PWT, the thermal constraint comes from electronic components and the electric motor but stays way lower than from thermic PWT (100 – 150°C max on the sealed areas). This has not a direct impact on the sealing strategy as there is also a lot of cold sealing type used for thermic PWT but having lower temperatures can sometimes allow to apply less pressure in the cooling systems. Indeed, in some cases, a pressure is applied in the circuit, firstly to have a good water circulation but also to be sure that the cooling liquid will not evaporate if the temperature is too high. Having lower temperature could permit in some case to decrease the pressure and so the mechanical solicitations.

2.2 Mechanical solicitations (batteries housing)

As said before, mechanical solicitations are not changing radically from E-PWT to TH-PWT. Mechanical solicitations mainly comes from the outside of the car and are vibrations and impacts.

Figure 27: Zoé battery system

But there are some differences concerning the sealing for battery system. Indeed, batteries on E-PWT are generally under the car and have big dimensions. Just above can be seen the Zoé and its battery (Figure 27). As the battery is very large and long and that it is located under the car, the sealing must be able to absorb several millimeters od deformations. Because of that, mechanical solicitations are higher for all the battery system. Moreover, battery system is generally supported by the car body that can be very flexible and this is even more deformations to be absorb by the sealing.

2.3 Electromagnetic compatibility (EMC) [7]

The major difference between E-PWT and TH-PWT are the EMI ((Electromagnetic interferences). Fossil fuelled vehicles also suffer from EMI. The ignition system, starter motor and switches cause broadband EMI and electronic devices cause narrowband EMI. But as compare to ICE (Internal Combustion Engine) vehicles, electric vehicles are combination of various subsystems and electronic components like battery, DC-DC converter, inverter, electric motor, high-power cables distributed around the vehicle and chargers, all these are working at high power and frequency levels which causes the emission of high-level low-frequency EMI.

EMC (Electromagnetic compatibility) of a device or equipment means its ability to not to be affected by electromagnetic field (EMF) and not to affect another systems operation with its EMF when it is operating in electromagnetic environment. In order to be sure that every component will not disturb others, some strategies are used. But from which principal component are the EMI coming from? (Figure 28)

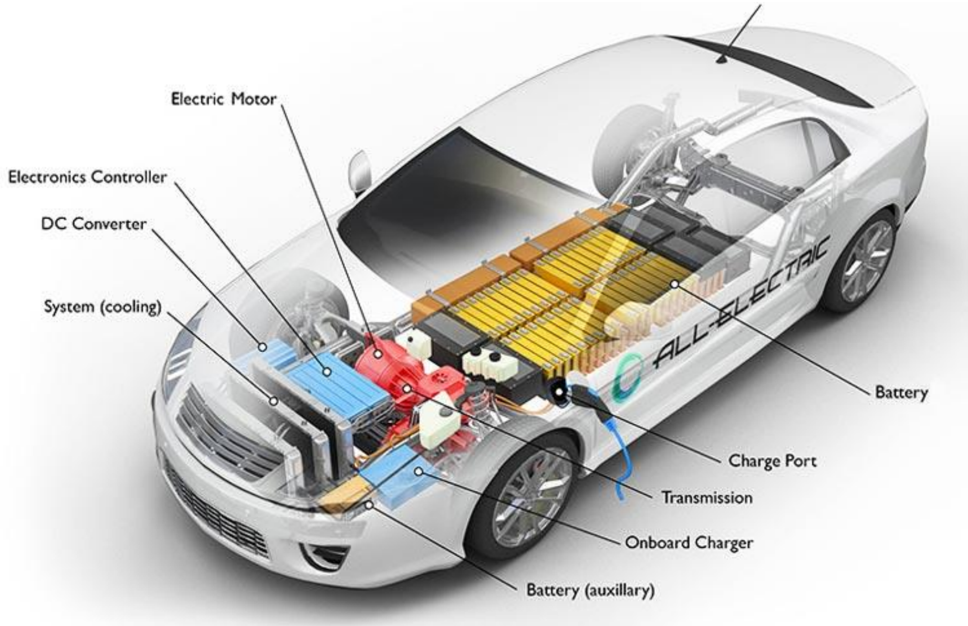


Figure 28: EMC leaks sources

- Power Converters are known to be the main source of electromagnetic interference within electric drive systems.
- Electric Motors which are operating at high power levels causes electromagnetic emissions and it act as path for EM noise through its impedance.
- As traction batteries are distributed, the currents in the batteries and in the interconnectors become a significant source for EMF emission
- Shielded and unshielded cables carrying high level current between various subsystems like battery to power converter, power converter to motor in the EV causes stronger magnetic fields. As available space in EV for wiring harness is limited, high voltage and low voltage cables placed near to each other causes electromagnetic interference between them.

- The battery chargers and the wireless charging facilities are the major external EMI sources. When wireless power technology is applied to charge the EV, a strong magnetic field in the range of several tens to hundreds of kilohertz produces to transfer several KWs to tens of KWs of power.

Nowadays with the advancement in technology, automobiles contain more electronic components. There is a large amount of electrical and electronic systems placed into a confined space in EV. This causes electromagnetic interferences. If EMC not maintained properly these systems may malfunction or even may fail to operate.

In order to limit the interactions between all the components and also to protect the driver from too high electromagnetic fields, some conception strategies are used. We will see those that can have an impact on sealing.

- The major rule is to be sure that the motor housing should be grounded to the chassis in order to minimize electric potential. That means sealing should not isolate some parts from the others.
- As steel has better shielding properties than aluminium, steel housing will be preferred for the motor.
- Some steel plates may be necessary close to sealing area in order to shield the sealing.

2.4 Safety problems/ electric continuity

Finally, for safety reasons, every part of the powertrains should be grounded to the chassis. If not, this would endanger user’s life is they would touch some parts not grounded. To do so, electric continuity should be maintained between parts and this adds sealing constraints.

Indeed, in all the sealing technologies used for TH-PWT, it is hard to evaluate those which are conductive or not. For example, it is difficult to know if FIPG sealing are conductive or not. Is the steel/steel contact sufficient or can we consider that screws are conductive enough be sure parts are not isolated? For now, these things are hard to anticipate and are generally evaluated during trials. We will see later which technology are generally used for which applications and try to make a state of the art concerning sealing technologies for E-PWT

3 Benchmarking

There are several objectives with this benchmark. The first one is to familiarize with sealing technologies and parts to seals for E-PWT and TH-PWT. Then it will be interesting to identify which sealing types are preferred for which interfaces. In a second time, it would be interesting to define some sizing criteria by sealing types and measure them for some applications.

3.1 Which gasket for which application

The type of gasket used depends on several parameters. It can change with the material of the part to seal, the complexity of the geometry, the operating temperatures, and the pressure constraints.

3.1.1 TH-PWT

The presentation will be divided by engine parts, the upper part, the lower part, the cylinder head part and the timing cover part. (Figure 29: engine division)

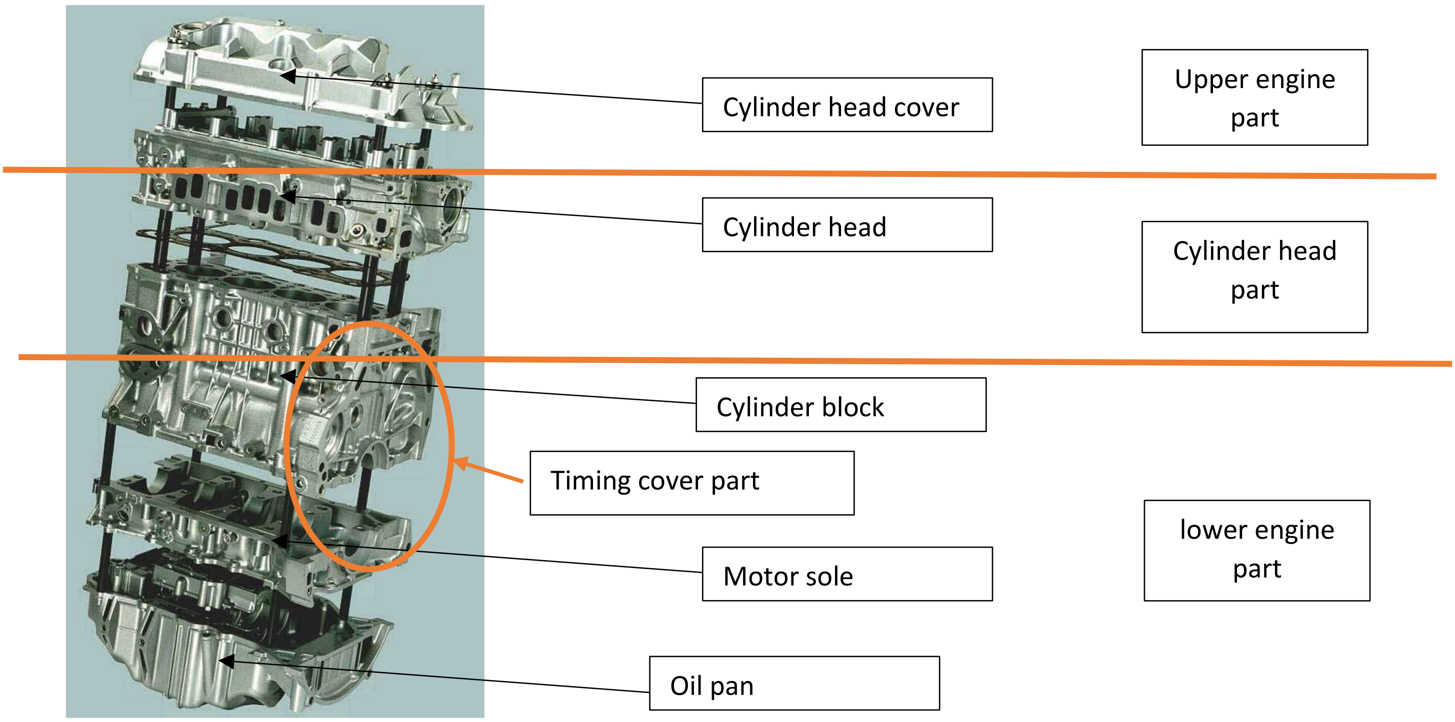


Figure 29: engine division

- Upper engine part

Concerning the cylinder head/ cylinder head cover: there are two possibilities

These last years, cylinder head cover used to be in metal alloy. In this case, the preferred sealing type is FIPG. For plastic cover, the tendency is to use gasket in a groove. Indeed, the use of FIPG on plastic parts is not advised. The first reason to this choice is the lack of adhesion of the FIPG on plastic/polyamide. There are more risks of non-cohesive rupture if the silicon adheres less with the plastic cover. A non-cohesive rupture is synonym of leakage, what we absolutely want to avoid.

Moreover, plastic parts have not the same mechanic properties as aluminium. Plastic parts are less stiff than aluminium parts and are more susceptible to creep in time. That is why it's not common to seal plastic parts with FIPG. Instead, elastomer gasket in groove are preferred.

Usually, the intake manifold is connected to the throttle body and the cylinder head with elastomer gasket in a groove. Indeed, the intake manifold often is in plastic and that's why FIPG are not preferred.

Due to the high temperatures in exhaust system the preferred seals are metallic gasket. Indeed, exhaust gases entering and leaving the manifold can reach 900 °C.

- Cylinder head

Cylinder head sealing are considered as hot sealing. Cylinder head gasket are very specific. Those are flat metallic gaskets and they are used to seal the cylinder head and the cylinder block. Those gaskets are quite complex and will not be detailed here as we want to stay focus on cold sealing.

- Lower engine part

For lower engine part, elastomer gasket in groove are preferred in case of plastic parts or for the water pump sealing. The use of FIPG is not recommended on the water pump function, in order to avoid any risk of pollution of the circuits by the sealing product (particle detachment). For the other parts (sole, oil pump, cylinder block) in aluminum/steel alloy, FIPG are preferred for economic and simplicity reasons.

For timing gear cover, most of the time FIPG is preferred. Cover geometries are often complex, that's why elastomer gasket in groove are not easy to use in this case (gasket rigidity problem for assembly feasibility).

3.1.2 E-PWT

This part on E-PWT will be less detailed than the previous one because as E-PWT are more recent, less engine has been disassembled so it is more difficult to get a global vision of what sealing technologies are used or not and how they are used. This part will be a kind of state of the art concerning E-PWT sealings.

Sealing application for E-PWT can be divided in 3 categories. The sealing of the motor housing, the sealing for the electronics components (HV charger, Power inverter) and the sealing for the battery system.

- Motor housing:

Three different sealing technologies has been identified to seal motor housing. FIPG, elastomer gasket in a groove and rubber coated flat metallic gaskets. These types of sealings are those used for TH-PWT. What could be interesting is to observe the difference of sealing design between TH-PWT and E-PWT knowing the different solicitations between them. A small study will be detailed in the next part to try to see some differences.

- Electronic housing:

Concerning sealings on electronic housings, technology used depends on the application and the geometry. For complex geometry, flat coated metallic gasket will be preferred but it is rarely used. For now, CIPG and FIPG are preferred but there are more and more elastomer gaskets in a groove used in electronic housing. The complexity here is to be sure to respect CEM criteria.

- Battery sealing

Battery systems need sealing between the two housings (lower and upper) containing the battery. Two sealing types are used. As batteries are quite big systems, elastomer gasket in a groove and flat metallic gasket are not advised.

The first sealing type is a cured gasket with elastic properties that is placed on the sealing flange by a robot. It demands a curing time of 30 min after application. The second one is an adhesive gasket placed on the sealing by a robot. It has not good elastic properties but doesn't need any curing time. As seen before, battery system is subject to some big mechanical solicitations and that's why elastic properties of the gasket are very important. Meanwhile it is quite binding to have to wait for the gasket to cure. The choice of gasket is made according to the demand and considering all these observations.

Having in mind a good idea of which gasket is used for which application, it will be interesting to observe, analyse and define which are the sizing parameters for sealing technologies that will be analysed or used in the after in the report.

3.2 Criteria definition and measurement

3.2.1 Method

As there is no access to physic parts or 3D model for competitors' engines, access to photos on a database (A2mac1) will be used. Some photos of disassembled engine have a scale distance. Those photos allow to measure some interesting parameters on excel. Annex 1 shows an example of measurement.

3.2.1.1 Engines choice

We choose to look at recent engine from Renault's main competitors. Indeed, looking at recent engine we are more likely to look at engine with corrected issues.

Moreover, at first, we choose to study thermic powertrains because for now there is a lot more experience on thermic powertrains than there is on electric powertrains. Electric powertrains are generally new engines with less feedback and their sealing strategies are usually inspired from thermic powertrains as the solicitations are not so different.

Two sealing types will be studied.

3.2.1.2 Sealing type choice

- Reasons for choosing to study FIPG:

As seen before, FIPG are the best quality/cost ratio for now. As it needs only delivery robots, for each different part, the only change to do is the robot's program. It is the technology that is preferred in many cases because it is cheap and easy to use. But that is also a technology with some recurring problems. Measuring some parameters from competitors would be interesting.

Moreover, Nissan and Renault are trying to converge their recommendations. Nissan makes smaller sealing flange and bigger chamfer for FIPG sealing in order to reduce the weight of the engine parts. But by reducing the sealing flange length, we are concerned it would endanger the sealing efficiency. The goal was to try to evaluate the sealing strategy of our competitors in order to know if they have the same tendency.

Finally, FIPG is a sealing technology that will be analysed in the study case. It will permit to understand more its principle.

- Reason for choosing to study elastomer gasket in a groove:

The tendency nowadays is to replace metallic parts by plastic parts to gain weight. FIPG is not usually used on plastic. Indeed, plastics have a lower surface energy than metals and therefore lower adhesion capacities. But why focusing on plastic parts? Plastic has a young modulus closer from elastomer gasket than metallic parts. Plastics parts are more likely to bend before the gasket. If the part bend, it can open a way for the fluid to pass.

Moreover, for the same reason as FIPG, elastomer gasket in a groove is a sealing technology which will be used in the study case. Understanding its principle and identifying important criteria and design choice can be useful.

3.2.2 FIPG

3.2.2.1 Selection of several parts

At the beginning, we started to look at a large diversity of parts, oil pan, cylinder head cover, timing cover. But looking at all these parts, we decided to focus on timing cover because of its complex and various geometry. It is a good way to limit number of parts to measure and still have a good overview of the technology.

3.2.2.2 Parameters to measure

To understand the sealing principle of FIPG, the best way is to start from the functional demand.

Functional demand: the gasket is the obstacle between the fluid and the outside. In order to have an efficient obstacle we need:

- Need: A good pressure repartition on the sealing flange/ sufficient contact pressure
  *Criteria: Bolt span and overhang (*Figure 30*). If some bolt span length or overhang are too long, the distribution of the clamping pressure is less homogeneous. It means some parts may not be under enough pressure and more likely to leak.*

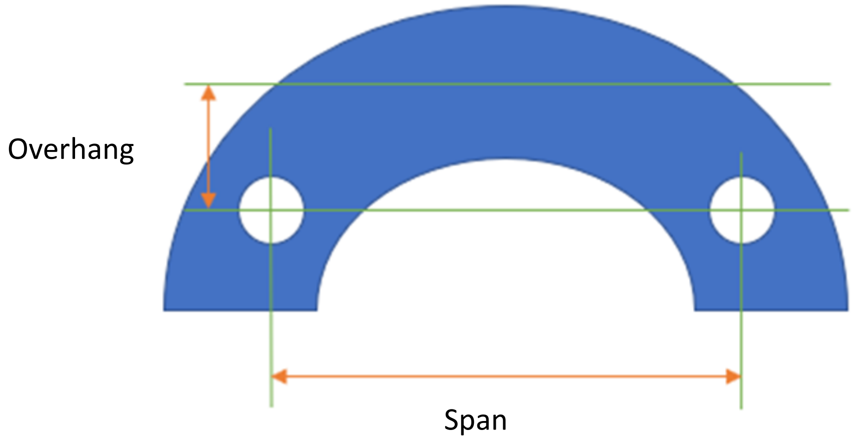


Figure 30: pressure repartition criteria

- Need: An obstacle large enough to maximize the chances that the fluid does not pass
  Criteria: Sealing flange length (Figure 30): The longer it is the more contact surface we have. The more contact surface we have, the more efficient is the sealing.

- Need: Capabilities to absorb parting line opening and a space to stock the excess of silicon. *Criteria: Chamfer length → spring reserve to absorb parting line opening, space to stock the excess of silicon*

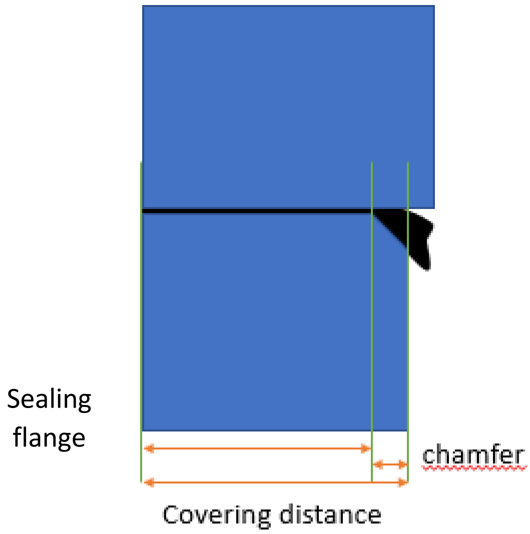


Figure 31: Chamfer and sealing flange

3.2.2.3 Interesting parameters we could not measure

Roughness would have been a good parameter to measure but it was not possible. the roughness of the parting line can influence the sealing efficiency of the assembly. The silicone first adheres to the part mechanically because of its roughness, this is the mechanical anchorage. The phenomenon is relatively simple: when the seal is removed, the silicone rushes into the roughness of the material thanks to its fluid behaviour. After cross-linking, the silicone can no longer come out of these roughnesses, which causes a kind of anchoring, thanks to the cohesion of the material (Figure 32). The rougher the substrate, the greater the tendency of the silicone to adhere (however, when the roughness become too high, this does not apply anymore). In addition, the effective surface area (actual surface area taking into account roughness) is greater when the roughness is high.

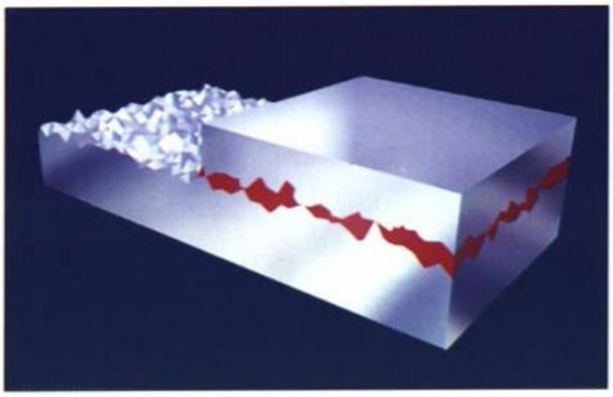
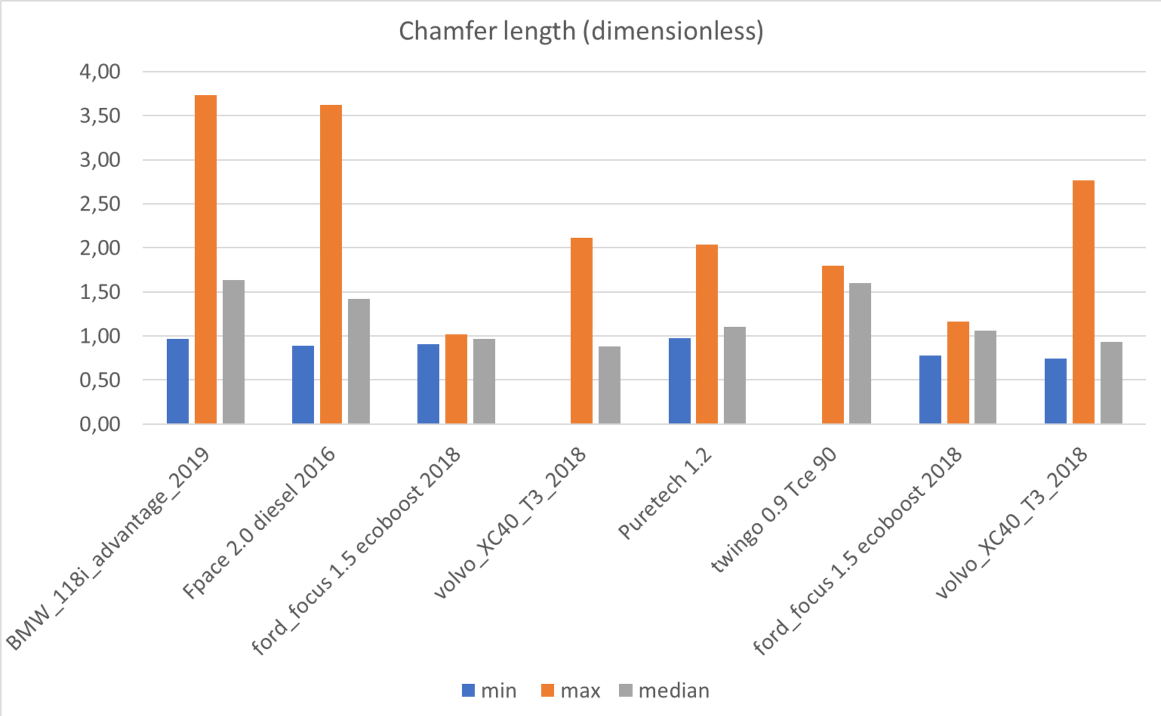


Figure 32: roughness and sealing

The silicon bead diameter would have been an interesting parameter to measure. It ensures that there will be enough silicon to form a barrier on the sealing flange but also that there will not be too much silicon in the chamfer. This diameter is generally validated by trials. It could be measured on photos as the assembly had already been done.

3.2.2.4 Results

For confidentiality purposes, the dimensions on the graph below are made dimensionless (Renault's recommendations are the references). Measurement confirmed that competitor's tendency is not to reduce sealing flange length and increase chamfer length but on the contrary, increase sealing flange length and decrease chamfer length. Indeed, on the measurements done, chamfer lengths are generally around 1 (it means that it is generally in accordance with Renault's recommendations. Sealing flange lengths are between 0,5 and 1,8 mm. These dimensions are quite like Renault's recommendations and confirm that there is no tendency to reduce sealing flange length to gain weight (Figure 33).



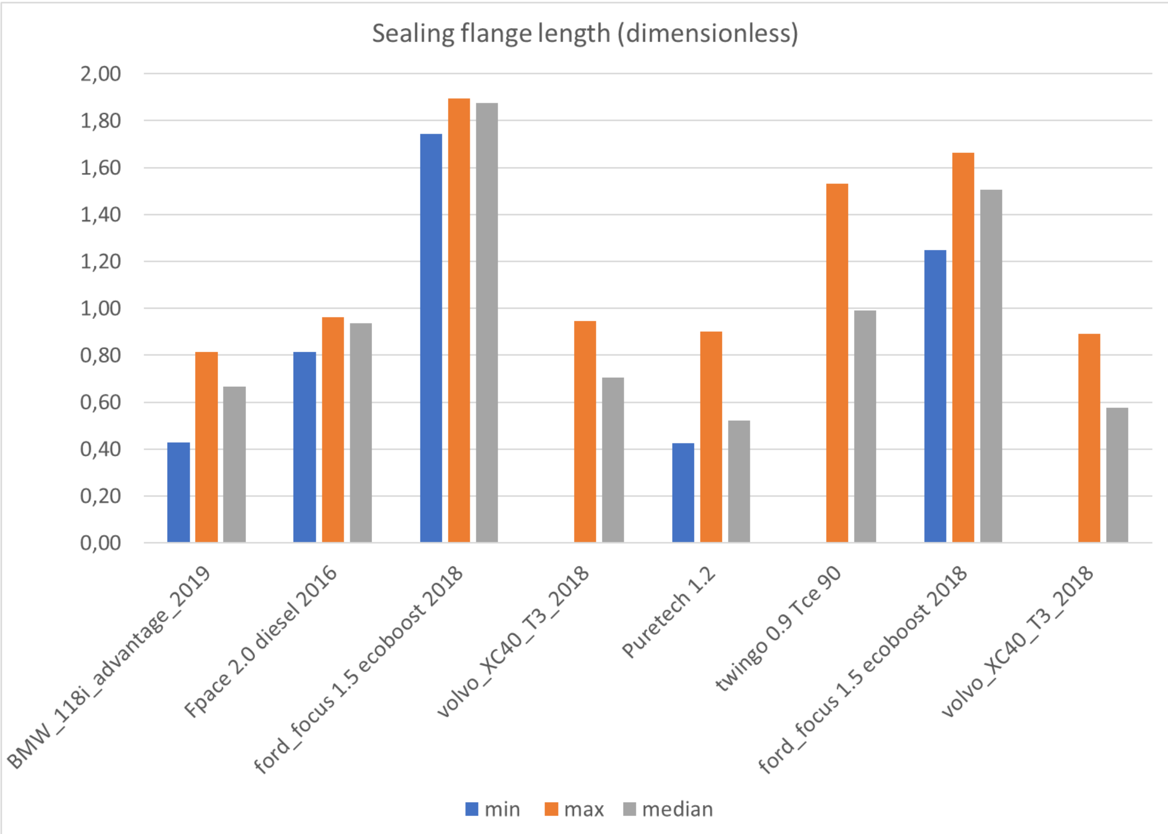


Figure 33: Chamfer length and sealing flange length for FIPG

The criteria used to check on overhang and bolt span values is the value "bolt span/overhang". Bolt span/overhang values have been measured for competitors and compared with Renault recommendations. Renault's recommendations fit with competitors designs.

3.2.3 Elastomer gasket in a groove

3.2.3.1 Selection of one specific part

Cylinder head cover are the most challenging parts using elastomer gasket in a groove. We decided to have a good overview to focus on them.

3.2.3.2 Parameters to measure

Functional demand: As it has been described in the bibliography part, sealing is ensured by the reaction force of the gasket after compression. In order to maximize the efficiency of the sealing, it needs:

- Creation of a contact pressure to form an obstacle for the fluid. The compression of the gasket creates a reaction force from the gasket. The compression ratio criterion is used. In order to respect it, two criteria are important:
    - *i) The gasket height, Hi*
    - *ii) The groove height, Hf (*Figure 34*)*

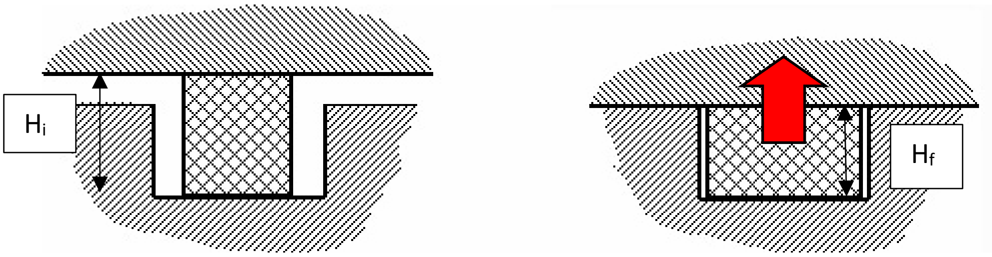


Figure 34: Gasket and groove height

- Repartition of the contact pressure
    - *i) Bolt span and overhang*

  The principle is the same as FIPG

- Stability of the gasket in its groove.
    - *i) Gasket width and pips dimensions*
    - *ii) Distance between pips*
    - *iii) Groove width*

  Ensuring gasket stability in its groove is ensuring that the gasket will not slip during the assembly process or after under operating conditions. For that the groove and gasket width are sizing parameters and the addition of pips can help to stabilize the gasket.

- Assembly feasibility. The more pips (Figure 34) there are the more friction there is to assemble the gasket in its groove. If there are too much pips, the gasket is too stiff and hard to put in its groove but if there are no pips, the gasket is hard to put in its groove because of bending, torsion and buckling.
    - *i) Pips dimensions*
    - *iii) Distance between gadroons*

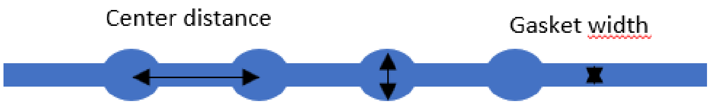


Figure 35: elastomer gasket in groove stability criteria

- Limit the bending of the cover
    - *i) Cover geometry: stiffness of the cover. If the cover is not stiff enough the clamping can deform it and endanger the sealing.*

3.2.3.3 Interesting parameters we could not measure

As for FIPG, there are some parameters which would have been interesting to measure but could be measured because it cannot be on 2D photos.

Groove height: could have permitted us to estimate the compression ratio. The photos are in 2D and taken from the top.

Gasket hardness: This parameter can change the contact pressure at iso-geometry.

Gasket surface: to estimate the confinement ratio. As geometry are often complex, sections cannot be measured on 2D photos.

Gasket stiffness: The stiffness influences the gasket stability in its groove. The stiffer it is, the more stable it will be in the groove.

3.2.3.4 Results

- We confirmed that almost nobody uses FIPG with plastic parts or if so, just for oil pan.
- Try to observe if there is a tendency for some dimensions:

<u>Stability</u>: No real tendency for the distance between gadroons (it depends on the geometry of the gasket and if the gasket surrounds fixation holes (Figure 37). In that case it seems the stability is partly insured by the fixation holes (Mercedes and Peugeot)

Figure 37: Fixations holes surrounded by the gasket

Figure 36: Fixations holes not surrounded by the gasket

Gasket geometry: a lot of similarities concerning gasket width, gasket height and gadroons width (respectively [2.5; 3.5], [8;9], [4;6] in mm). This gives us an order of magnitude of gasket dimensions for this application and these parts dimensions.

- It is difficult to characterize the assembly feasibility of the gasket in its groove. But what can be said is that usually, gasket that are very thin have more pips in order to increase the stiffness (a lack of stiffness can increase the difficulties to assemble the gasket). The Ford, Kia and Mazda gasket have a lot of pips because they have not a very robust initial geometry.
- The sealing flange height have been measured but there is no real tendency. It is between 6 and 12 mm depending on cover's geometry.

3.2.4 Comparison between E-PWT sealing technologies

The objective here is to try to identify differences between the different sealing types for E-PWT and explain them with CEM constraints. In order to do that, three different sealing technologies will be analysed (elastomer gasket in a grove, FIPG and coated flat metallic gasket). To get an idea of parameters differences three motor housings have been selected (one sealing technology by motor housing) with similar global dimensions to get comparable data.

- Parameters to measure:

As said before, the objective is to see the influence of CEM constraints on the sealing design. As the important criteria is the metal/metal contact between the two motor housings, bolt span and sealing flange length will be measured. Indeed, a part of the metal/metal contact is ensured by the screws. We will try to compare sealing geometry between TH-PWT and E-PWT and between each sealing type for E-PWT.

- Results: (Results have been made dimensionless for confidentiality purposes, the goal here is to compare dimensions)

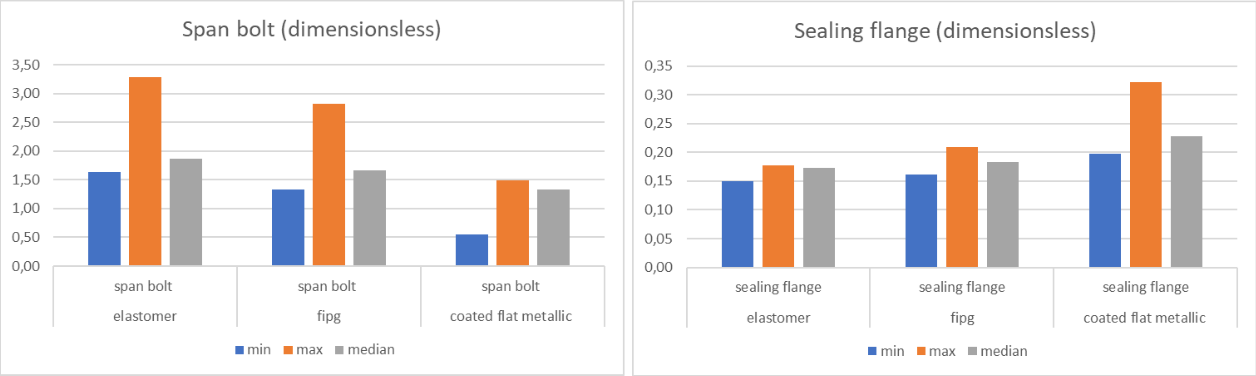


Figure 38: span bolt and sealing flange measurement for motor housings (E-PWT)

Two remarks can be done:

- Firstly, there is no big differences between sealing geometry for E-PWT and TH-PWT. Sealing flange and span bolt values are quite similar for FIPG and elastomer gasket in a groove.
- Comparison between each sealing type for motor housings indicates that span bolts tend to be longer for elastomer gasket in a groove (Figure 38). That means that for fixed dimensions, there are more screws on the sealing flange for coated flat metallic gasket. Indeed, for elastomer gasket, there is a metal/metal contact on the sealing flange around the gasket. But for coated metallic gasket, the metal/metal contact surface is limited because the gasket is rubber coated. The metal/metal contact is in this case ensured by the screws.

However, these observations are not sizing rules. Sizing recommendations concerning sealing flange design on E-PWT are still unclear. The EMC and equipotentiality is checked by trials but there are not clear sizing rules yet.

3.2.5 Results reliability

To confirm the validity of these results, it was important to be sure this method was reliable.

In order to confirm that, the easier way was to compare measurement from excel with measurement from a 3D-model from the same engine. An engine from Renault was chosen because photos and 3D-models were accessible. Measurement were made for FIPG sealing type on bolt span and sealing flange length.

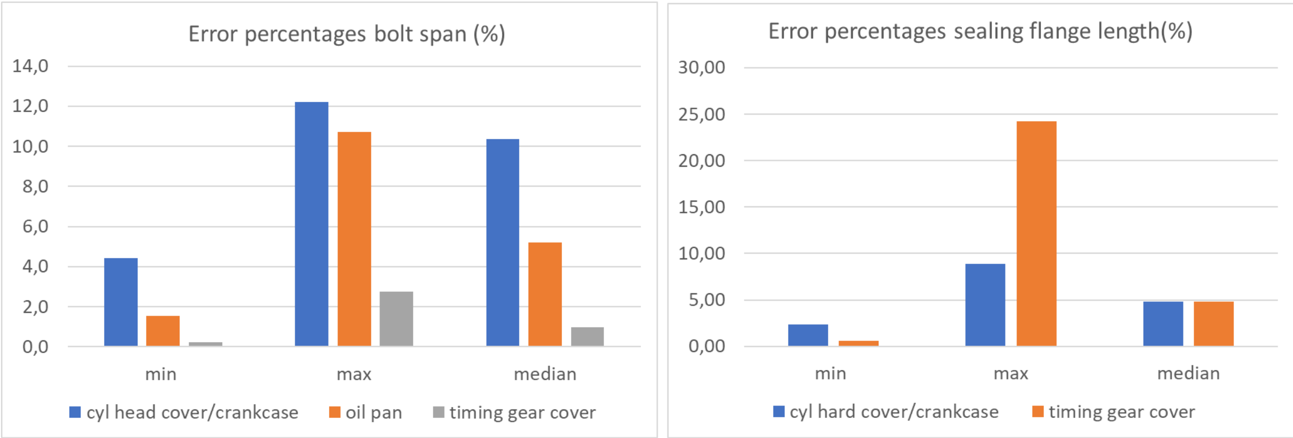


Figure 39: error percentages for benchmarking measurements

In general, the error is not higher than 15% (Figure 39). That confirm that we can exploit the results knowing they are reliable. Some values have more than 100% error but only one or two values. These errors are partly due to perspective. Some values have not been measured because of this perspective error.

3.2.6 Conclusion

This benchmarking part permitted to get a better understanding of each sealing type and more particularly elastomer gasket in a groove and FIPG. Starting from the functional demand associated with each sealing types, sizing criteria has been identified.

For FIPG, it confirmed the importance of having a sufficient sealing flange length. Even if reducing it would be a weight gain, it would also make the sealing less robust. For elastomer gasket in a groove, it will be helpful to understand the usefulness of each gasket parameters for the study case. Finally, for E-PWT, it is a kind of state of the art for sealing technologies. An interesting thing would be to write sizing recommendations for E-PWT sealings thanks to studies and trials.

All this part will be useful to do the next part of the report which is the study case.

4 Study case: PEC 5A gen 3

In this part, a defective sealed assembly will be studied. First the objective is to identify possible root causes for the problem, and then a solution will be proposed.

4.1 Introduction

The Power Electric Controller (PEC) is a generation 3 device that fulfils all the functions of a power electronics and has a dual role in the E-PWT:

- battery charging (charger mode)
- control of the electric motor (traction mode or regenerative braking).

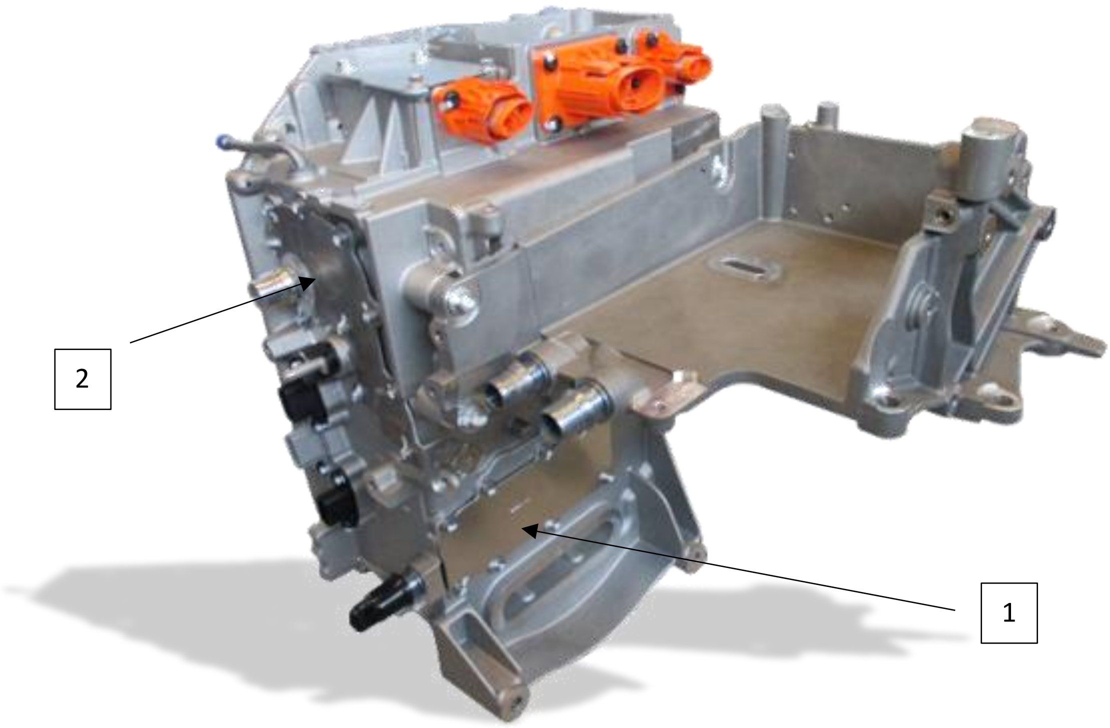


Figure 40: PEC 5A gen 3

Also integrated in the PEC is the DC/DC which supplies the 14V functions and recharges the 14V battery.

On this PEC, there is a cooling system to cool down all the electronic components. This cooling system is divided in two parts which can be seen just above. The first one is the bigger one. It can be seen just beside on the two photos and on Figure 41.

Figure 41: first part of cooling system for PEC

The second one is smaller and can be seen on Figure 40 and Figure 42.

Figure 42: second part of cooling system for PEC

- Geometry choice:

On the two parts, the geometry is conceived trying to maximize the heat exchange surface to cool down more efficiently.

- Assembly choice:

Those two parts are composed of a carter and an aluminium plate assembled with M5 screws clamped with 6,5 +/- 15% Nm torque. Usually, for M5 screws the recommended torque is 4,5 Nm but, in this case, some silicon can be found under the screw's head during clamping. Clamping screws with silicon under its head can weaken the clamping. That is why the torque 6,5 Nm is applied. The cover is in aluminium because it is a good compromise between stiffness and weight.

- Sizing constraints for sealing:

There is a 1,4 bar pressure in the circuit. This value comes from thermic powertrains cooling system. Indeed, with the temperatures thermic powertrains can reach, in order to avoid boiling the coolant liquid, a pressure is applied. However, temperatures are lower for electric powertrains. Temperatures can reach more than 100 °C in some cases but that would be for one faulty electronic component and it would not resist this temperature. This pressure is not very useful in this case and is the proof that thermic and electric solicitations can be different concerning sealing.

Moreover, a maximum parting line opening of 50 µm has to be respected.

- Sealing choice:

Usually FIPG are not used for circuit under pressure but in this case, to seal the assembly, FIPG is used between the two parts. It is a cheap and easy way to seal an aluminium and metal part.

4.2 Problematic

In 2016, several vehicles experienced leakage on the red circle on the Figure 43.

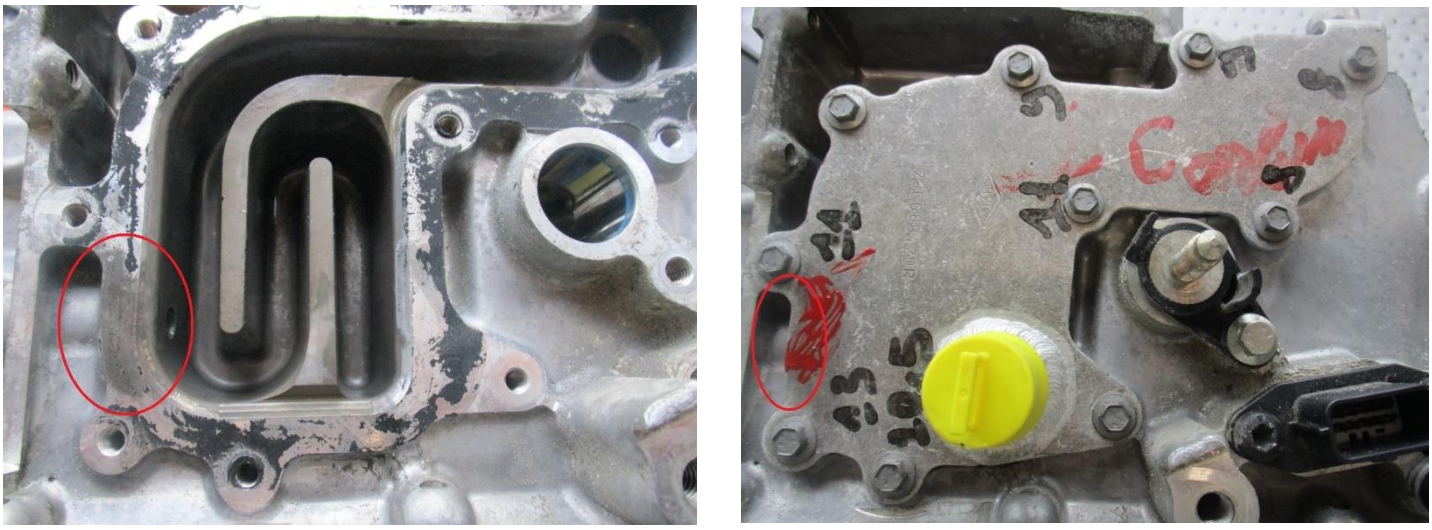


Figure 43: PEC leak

Since then some studies and modifications have been done to try to fix the problem. The next part will list everything that has been observed and done to fix the problem. At the end, some strategies to explain the problem will be proposed and next parts will detail them.

4.3 Failure tree

In order to find the root cause of the problem, an efficient solution is to make a failure tree (Figure 44). This consist of identifying all the possible root causes. This can be divided in 2 categories, product, and process. The possible root causes from a product point of view can be geometric or related to the state of surface definition. Indeed, a geometry not robust enough could allow a leak. From a process point of view a lot of root causes can be identified (clamping, state of surface, operator mishandling, position of the silicon cord).

All these possible root causes must be investigated in order to find what could cause the leak.

Failure	Causes	Sub-causes		Effects	Checking tools
Water leakage from FIPG surface (cold sealing) --> parting line opening	Geometry	sealing flange geometry	chamfer length too short	insufficient elastic reserve / particle detachment	measurement on 3D model/ comparaison with Renault's recommendations
			sealing flange length too short	weaker barrier for the fluid	
			span bolt too long	bad clamping force repartition	
			overhang too long		
			covering length tolerances	covering length too short	Tolerance stack up
		cover geometry	cover width too low	high deformations	Simulations on Catia
		bad roughness definition of the sealing flange	R5-25, Rx30	adhesive efficiency	roughness definition
		flatness			Tolerance stack up
	process/ assembly	curing time before tests			Process analysis
		Surface pollution	detergent residue	adhesion problem	
		scratch on the sealing flange			
		bad clamping	clamping order	cover deformations	
			clamping torque	clamping force repartition	
		roughness of the surface not respecting definition		adhesive problem	Roughness measurement
		delivery path	bad positioning of the FIPG cord	covering length too short	Process analysis
		operator mishandling	Difficulties during the assembly	sealing deterioration	

Figure 44 : Failure tree

4.3.1 Product

4.3.1.1 Roughness

The first possible root cause investigated by Renault has been the surface roughness of the carter (on the sealing flange). Indeed, the roughness definition gives a maximum roughness but not a minimum roughness as advised in that case. The definition is a maximum roughness of 25 µm but a minimum roughness of 5 µm should also be defined. Indeed, as seen before, if the roughness is to low, the silicon has less possibilities to anchor in the asperities of the surface.

To analyse the roughness, two criteria can be distinguished, R and Rx. R is the average depth of roughness patterns and Rx the maximum depth of roughness patterns.

The measurement performed on the surface revealed a surface roughness R of approximately 0.8µm and a Rx of approximately 4.5 µm. These values are much lower that the recommendations.

But this would not explain why the leak would be localized on only a part of the sealing flange. What have been noticed in addition with this problem is that the machining tool passes two times on some parts of the sealing flange. The part that leaks is one of them. This can be seen on the machining path below (Figure 45).

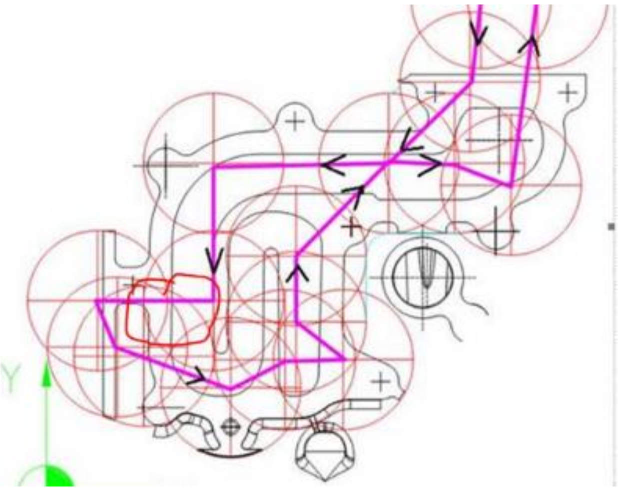


Figure 45: machining path PEC

The problem is that from a process point of view, it is quite difficult to machine a part respecting a minimal roughness.

As mentioned earlier, the cooling system is divided in two parts. But leaks have been observed only on the smaller part. As the roughness definition is the same for both parts, we could have expected to observe leaks on the bigger part of the cooling system if this is only a roughness problem.

So even if this could explain partly the leak, the other possible root cause from the tree must be investigated to evaluate if it could be a combination of several problems.

4.3.1.2 Sealing flange geometry

To compare the two parts of the cooling system and to try to find out why one leaks and not the other, the sealing flange geometry can be analysed. The sealing flange geometry is, has seen before, directly related to the sealing efficiency. If some differences can be found between them, then it can be a possible root cause for the leak. Moreover, measuring both geometries will be a way to check if it respects Renault's recommendations.

In order to measure precisely these geometries, the measurements will be performed on 3D model. The parameters to measure are the parameters that have been described in the bench (Bolt span, overhang, chamfer length and sealing flange length).

Bolt span and overhang are easy to measure as shown beside Figure 46).

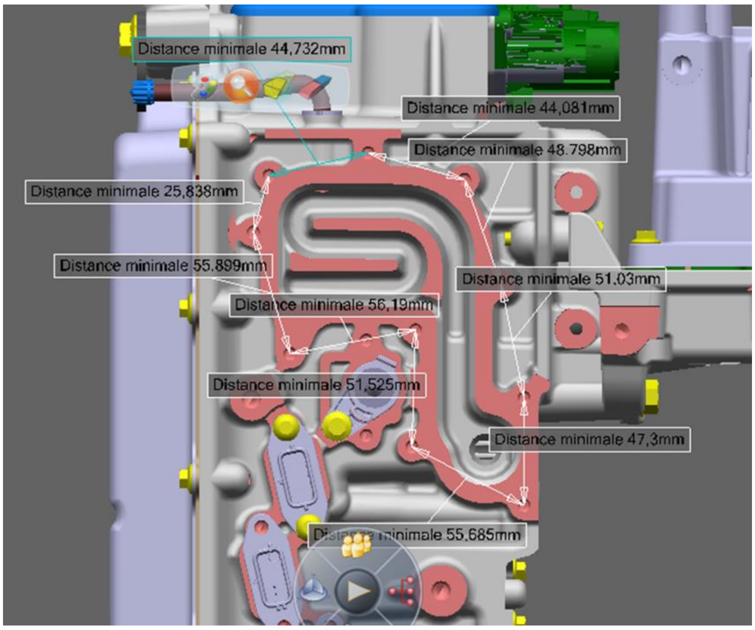


Figure 46: bolt span measurement, PEC

However, the sealing flange length and the chamfer length are more difficult to measure. Indeed, it is not constant between two fixations holes, so the choice has been made to measure it where the sealing flange seems to be the shortest. An example is shown on Figure 47.

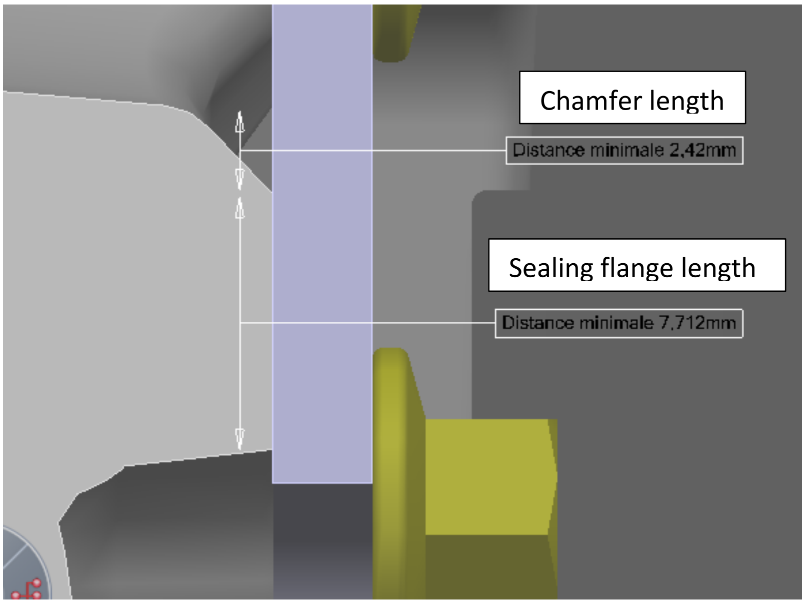


Figure 47: sealing flange and chamfer measurement: PEC

Those measurements were made between each fixation's holes.

Both parts of the cooling system have been measured with the same method. As their geometries are not exactly similar, comparing them will be interesting but will not help to conclude about the robustness of the geometry with its solicitations. Table comparing minimum, median and maximum for each parameter can be found below (Figure 48).

	small cover				big cover			
(mm)	bolt span	covering length	chamfer	overhang	bolt span	covering length	chamfer	overhang
min	25,83	7,66	2,29	0,00	42,65	6,89	2,84	2,80
median	49,92	7,93	2,71	4,58	61,47	7,02	2,98	7,21
max	56,19	9,50	3,02	13,48	71,83	8,10	4,76	9,40

Figure 48: comparison between small and large cover, PEC

What can be observed is that bolt span and overhang are shorter for the small cover geometry and that is synonym of a better pressure repartition on the sealing flange. One overhang value is longer, but it is not the on the part of the sealing flange that leaks. In general, covering length and chamfer length are quite similar. Moreover, all these parameters respect Renault's recommendations.

This study shows that the geometry is quite robust. However, it would be interesting to observe the parting line opening when circuit pressure is applied, and screws are tightened.

4.3.1.3 Parting line opening [8]

Simulations on Catia V6 can be performed to analyse the parting line opening under operating conditions. First the model used for the simulation will be described, and then results will be compared to Renault's recommendations.

Model

<u>Parts:</u>

The model is composed of two parts, the cover, and the housing. The housing has been cut to reduce simulations time. Model's parts can be seen on Figure 49.

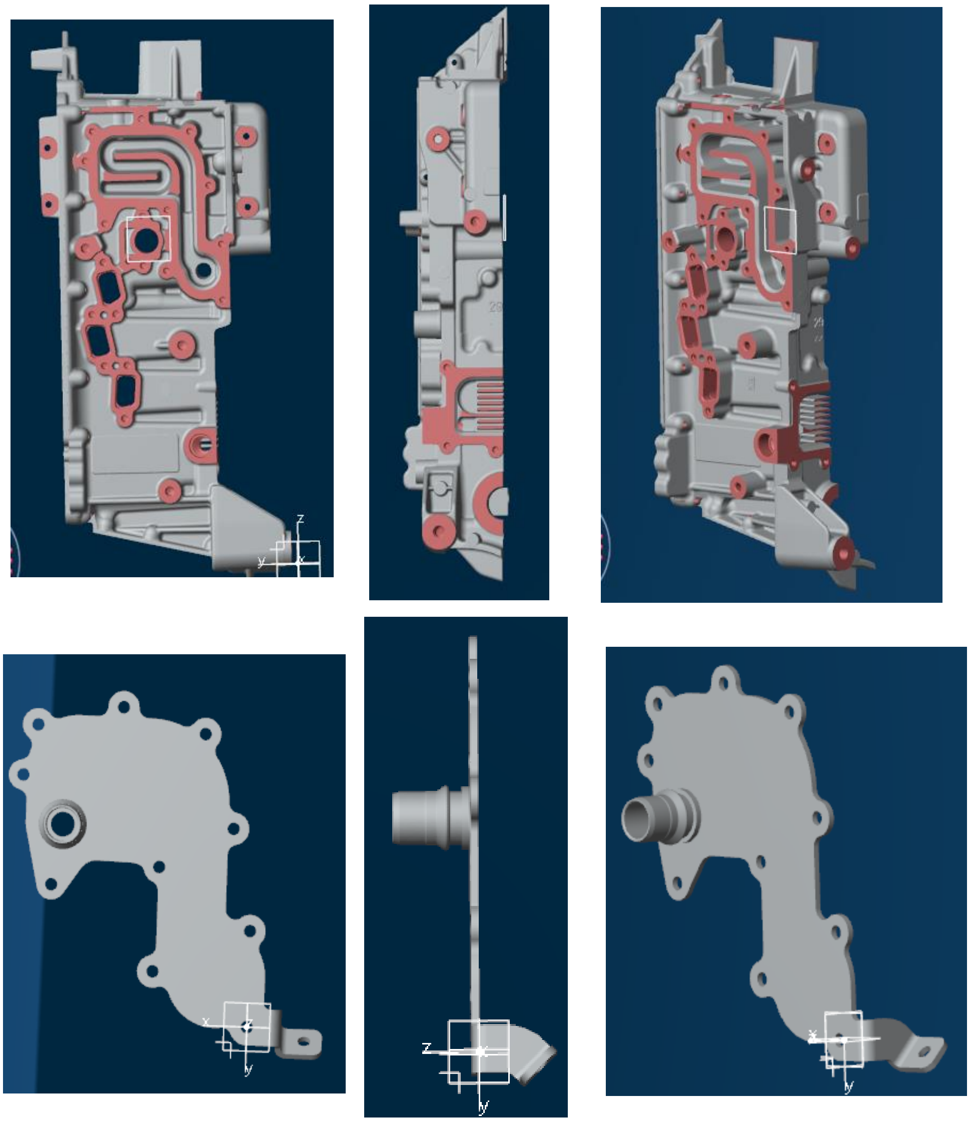


Figure 49: model's parts

Bindings:

In order to model the screw, a tension force and a stiffness are applied between the carter and the cover. The tension force of M5 screws is known but stiffness of M5 screws is unknown and simulations has been performed to find it (Figure 50). The tension force is between 4400 N and 7620 N. Maximum and minimum tension forces will be tested to see its impact on parting line opening. Concerning stiffness, simulations has been performed on one screw to find its stiffness. In order to do that, a force of 10 N was applied under the screw's head and the screw base has been fixed.

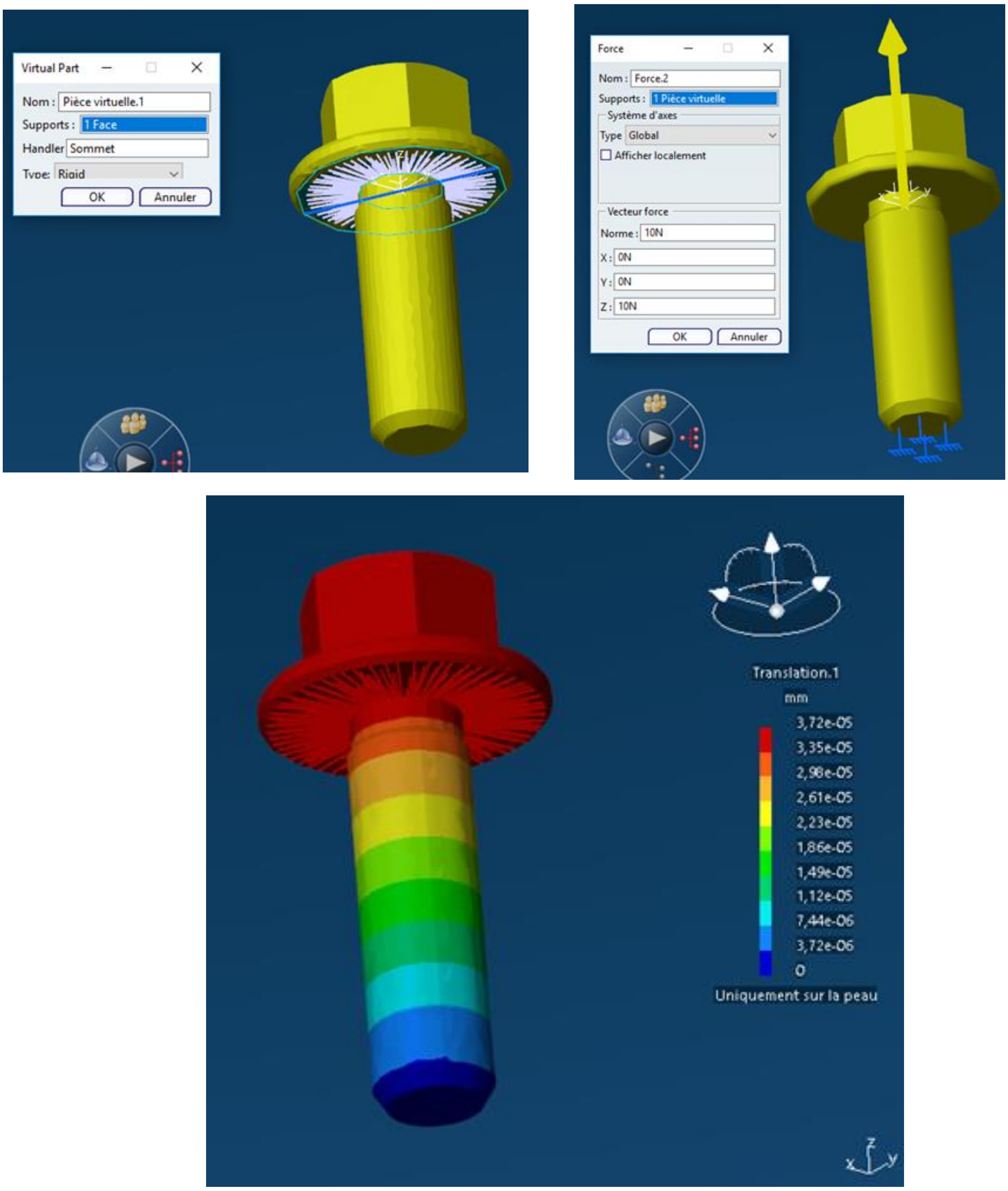


Figure 50: Screw stiffness simulations

The head displacement is observed and then by dividing the force by the displacement, stiffness coefficient is obtained. The stiffness coefficient is 2.2×10^8 N/m. In the software 3 different stiffnesses are asked for the 3 axes, but the two radial stiffnesses are not important as the solicitation is a traction force on screws (Figure 51).

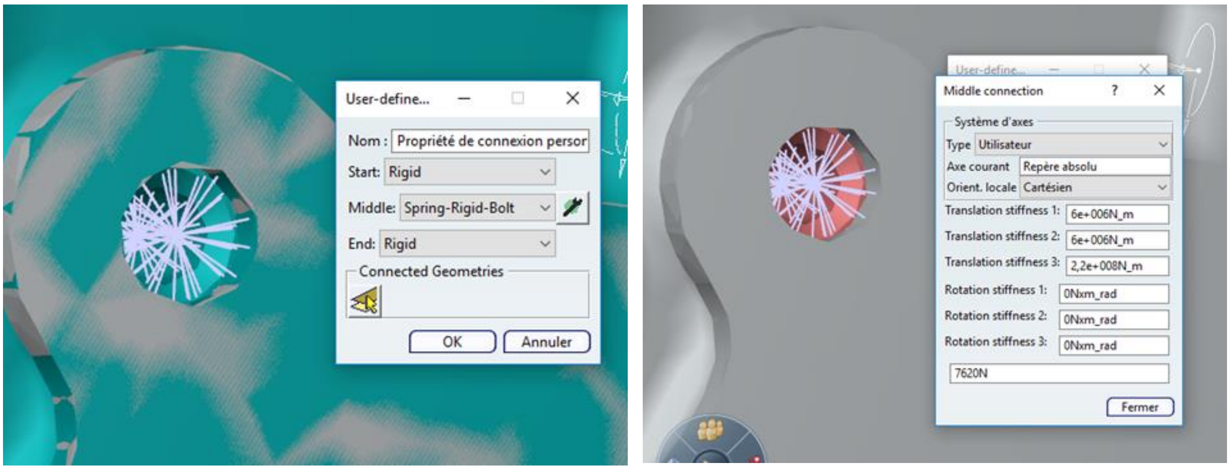


Figure 51: screw modelisation

Contact surface modeling:

Then the interaction between the two surfaces (sealing flange for the cover and the housing) needs to be defined. The two surfaces are in contact and a friction coefficient of 0.3 is applied (friction coefficient between two aluminium alloy parts).

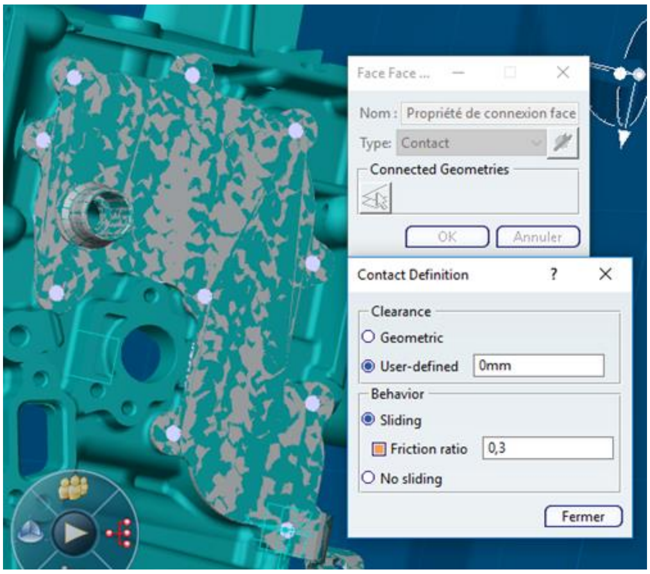


Figure 52: contact surface modelisation

Material affectation:

Here the material is an aluminium alloy for both parts. The Young modulus, Poisson coefficient and density of this material can be found below.

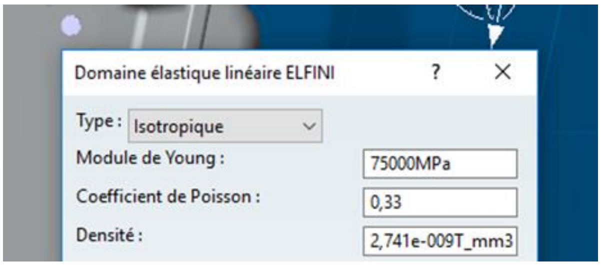


Figure 53: material used in the model

Mesh definition:

The meshing is performed with tetrahedral elements. The mesh is more precise for the cover because the cover displacement and deformations will be analysed. On the right below is the mesh definition for the housing and on the left the mesh definition for the cover (Figure 54).

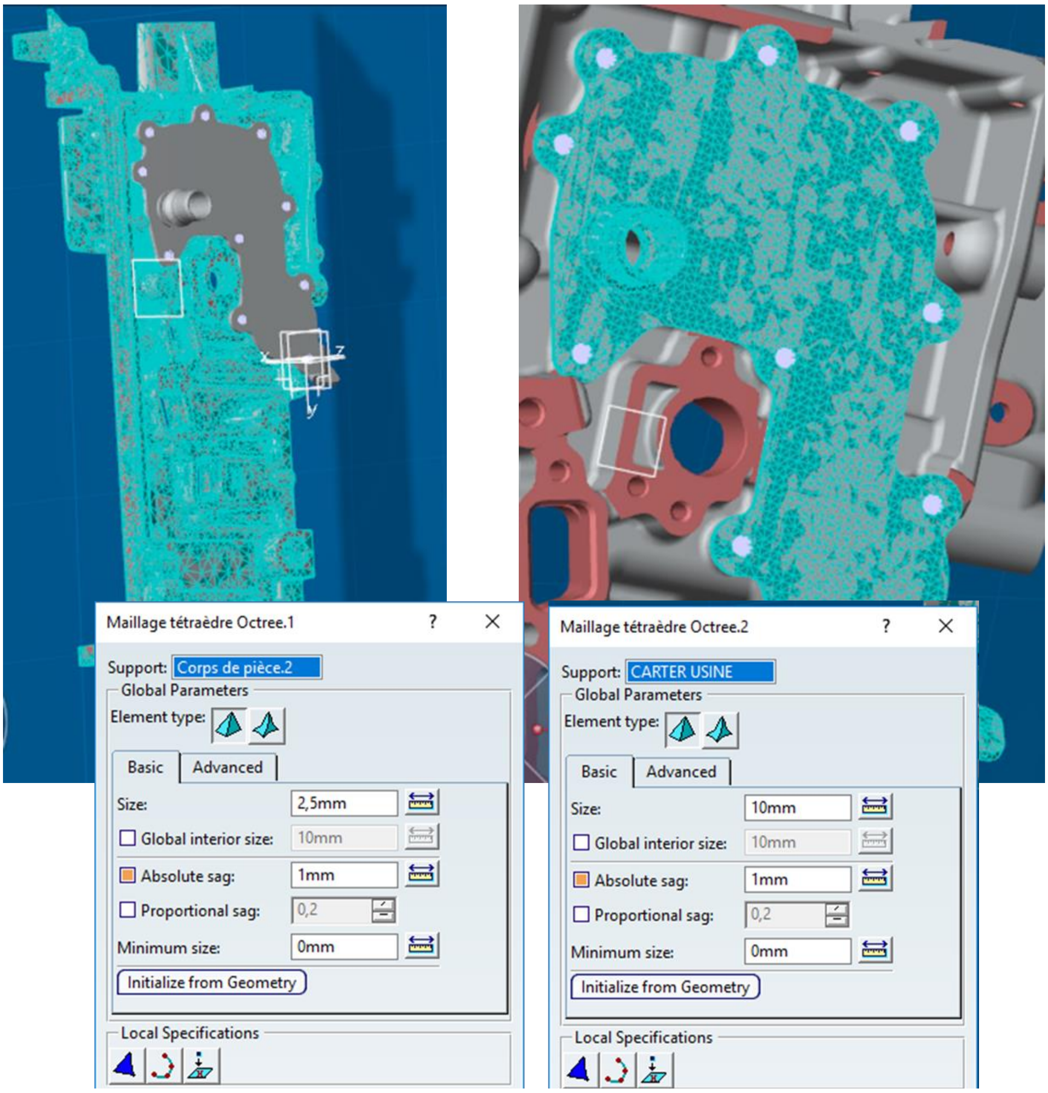


Figure 54: housing and cover mesh

A local mesh has been defined on the sealing flange in order to get more precise results on the parting line opening. Figure 55 shows on which surfaces the local mesh is applied.

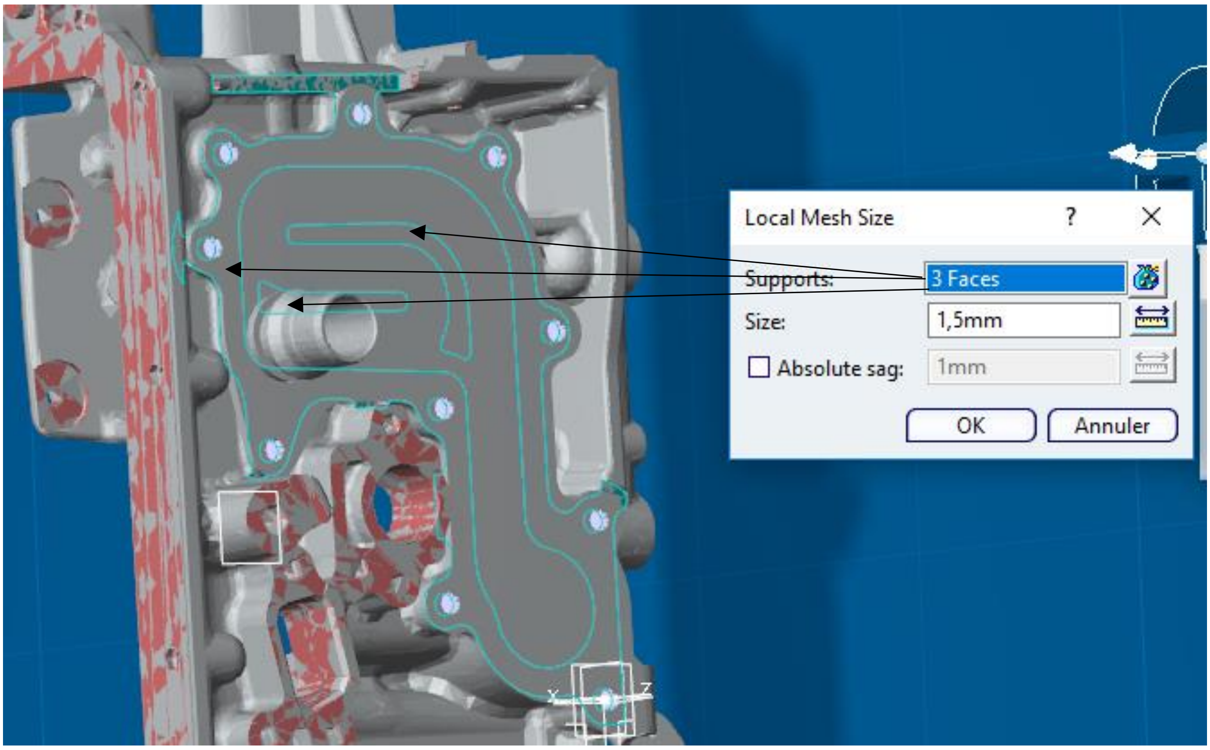


Figure 55: local mesh

Constraints and pressure:

The housing needs to be embbeded and a pressure of 1.4 bar is applied on the cover surface.

Figure 56: housing embedding

The pressure is applied on the inside surface of the cover where there is no contact between the cover and the housing

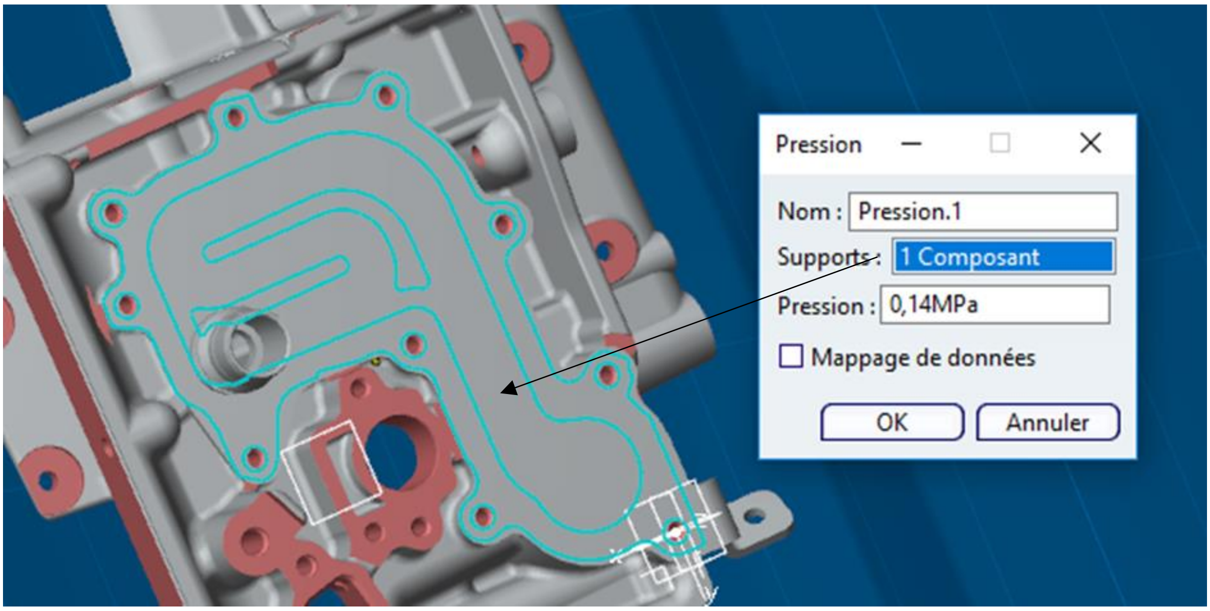


Figure 57: pressure modelisation

- Results:

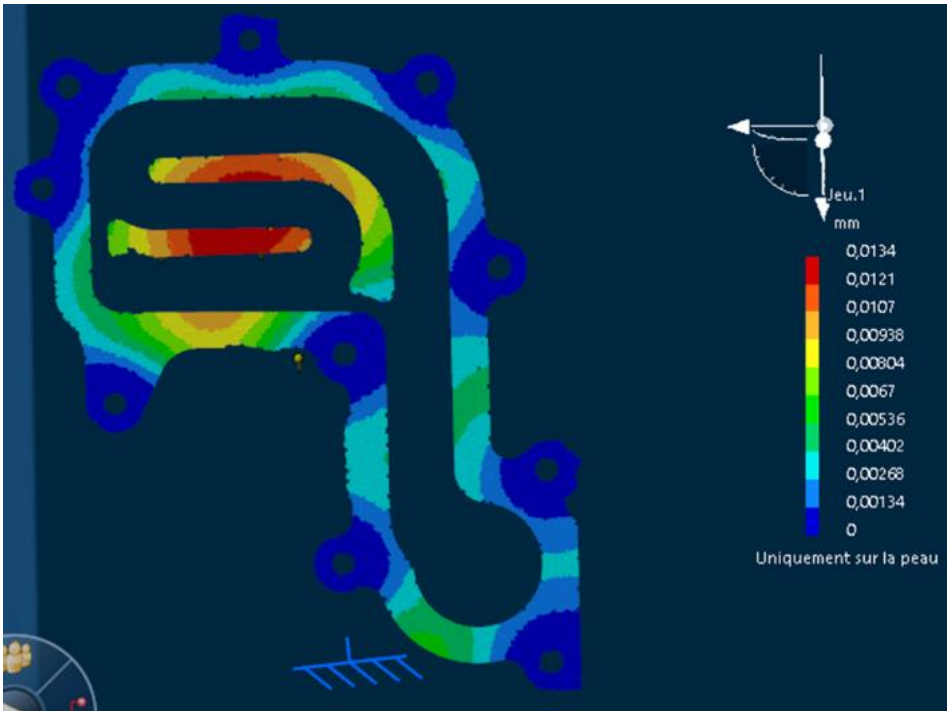


Figure 58: parting line opening for maximum screw tensions

Simulations have been made using different parameters for the screw tension and stiffness. Two simulations have been performed using the maximum and the minimum screw tension to know if this could be a source a problem. Results can be observed parting line opening for minimum and maximum of screw tension on Figure 58 and Figure 59.

Figure 59: parting line opening for minimum screw tensions

By analysing these results, several conclusions can be made. First, results show that a higher screw tension results in the small increase of the parting line opening. Indeed, by clamping harder, the cover deforms more, and it creates a higher parting line opening. However, these parting line opening stay way lower from the limit value of 50µm recommended. Finally, the maximum parting line opening is not on the leaking area. These simulations permitted to conclude that the design is robust enough to resist to the mechanical solicitations.

From a product point of view most of the possible root cause has been investigated. The roughness definition could be a possible root cause to explain the leak. The geometry and the robustness of the sealing do not seem to be a problem has it respect the recommendations. The tolerances have not been investigated by lack of time but could be a cause if it reduces too much the covering surface.

4.3.2 Process

From a process point of view, answers come from those who are handling the assembly of the cooling system.

- Roughness: as described in the previous part, the roughness could be a problem as it is too low in comparison with the recommendations.
- Pollution on the surface: during and after all the machining, some cleaning and lubricating products are applied on the parts. These products can stay even after the assembly and endanger the sealing. Checking on this potential problem is not easy and has not been precisely investigated.
- Clamping torque: the residual torque has been checked and is high enough.
- Silicon cord positioning: the position of the cord on the sealing flange has been checked and validated.

- Curing time of the silicon: A sufficient curing time for the silicon before pressure application is not respected. This time to let the silicon cure is very important to get an efficient sealing. This recommendation is not respected. The sealing is validated if there is no leak during the trials.
- Mishandling, mis-assembly from the operator: It has been noticed than sometimes, operators used a hammer to assemble a water tube on the cover and that it could endanger the sealing has the silicon was not completely cured.

4.3.3 Conclusion

By investigating all these possible root cause, the conclusion is that the failure may have several causes. All is summarized in the Annex 2 with the full failure tree. First the too low surface roughness can be a source for a lack of adhesion of the silicon on the surface. Then some process possible causes can be identified. The time for letting the silicon cure is not respected before applying pressure and sometimes as the silicon is not cured enough, some assembly technics (hammer) can damage the sealing.

This study showed that there are several possible root causes and that playing with only one parameter may not be enough to fix the problem as this could be a combination of several causes. Indeed, to create an efficient sealing, a combination of several parameters and process must guide the sizing and assembly strategy and that is this combination that will ensure a good sealing.

In order to try to fix the problem, a new proposition will be done with a different sealing type.

4.4 Elastomer gasket in a groove sizing

First, the sealing type need to be chosen. Two sealing types could be used to replace FIPG. LEM and elastomer gasket in a groove can be used to conceive this sealing. As seen in the first part, LEM are more expensive but easier to use. Indeed, there is no need for a groove, it means less machining on the carter and an easier assembly. However, this technology is not the first choice yet because of the lack of experience. This technology is quite recent. Elastomer gasket in a groove is a technology which is often used for circuit under pressure and the sizing method and criteria are more detailed than for LEM. That is why elastomer gasket in a groove will be used to re-conceive the PEC sealing.

In order to conceive this sealing, it is important to begin by analysing the functional demand associated with the use of this sealing type. These sealing needs has already been detailed in the benchmarking part.

To perform the sizing, some gasket materials and parameters need to be defined.

4.4.1 Material

The material that will be used for the gasket will be an EPDM. Indeed, the temperature range of EPDM is adequate with the operative conditions and it is compatible with coolant water (Figure 60).

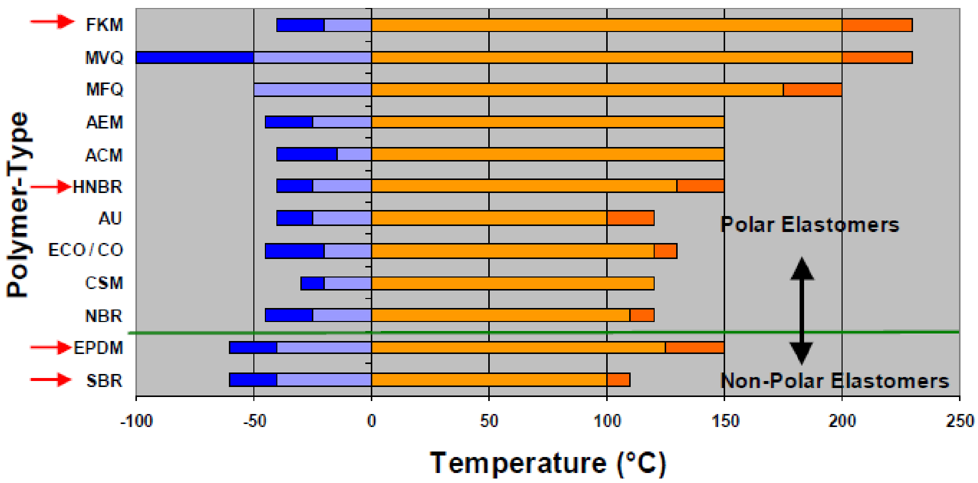


Figure 60: Temperature range by elastomer material

4.4.2 Parameters

First, the young modulus of the elastomer needs to be defined. Generally young modulus is not used for elastomer. The parameter used to define an elastomer is its hardness. To estimate the young modulus of an elastomer, Renault uses a curve that shows the young modulus as a function of the hardness. The hardness used for this type of application is between 65 and 70 IRHD.

The curve indicates a young modulus of 3.5 MPa for a 67 IRHD. This value will be used for the next calculations.

Then, a friction coefficient between the gasket and the aluminium cover needs to be defined. This friction coefficient is between 0.25 and 0.45. The value used for the calculations will be the one which is the most critical for our calculations.

Finally, a relaxation coefficient will be needed. The relaxation coefficient gives an indication about the capacity of the gasket to keep its contact pressure. Indeed, with the time, elastomer relaxes and the contact pressure decrease. This depends on some parameters like temperature or compression ratio. The relaxation coefficient is the percentage of initial pressure that is lost in time. For EPDM, the relaxation coefficient is between 0.2 and 0.4. These values have been found by searching in studies [9]. It is difficult to get relaxation values for elastomer and when needed these are often given by the suppliers. The value generally used by Renault for an EPDM is 0.25.

4.4.3 Contact pressure

First a sufficient contact pressure must be applied to form a barrier between the fluid and the exterior.

At first, we will not care about the section of the gasket. We will consider it as a rectangular gasket. This will permit us to perform the first sizing calculations in order to approximate the needed contact pressure.

But why do we need a contact pressure between the gasket and the cover and how to keep it ?

The contact pressure forms the barrier between the inside and the outside for the fluid. What are the problems that could suppress this contact pressure?

- A slippage of the gasket in the groove because of the circuit pressure.
- A slippage of the gasket in the groove because of the lack of roughness (influence on the friction coefficient)

The first possibility can be studied with Coulomb's laws but the second one will be more difficult to study as we have not data to link the friction coefficient to the roughness of the surface.

In order to study the slippage of the gasket we will use a very simple model. Hg and Lg are respectively the groove's height and groove's width. For now, we will take a groove's height of 3.6 mm and width of 4 mm. (H = 3.6mm and L =4mm). The strategy is to impose groove dimensions that can fit on the existing sealing flange and to design the gasket in this groove. Figure 61 shows the model to study the slippage. This model will give us a minimum gasket height to respect to be sure the gasket does not slip on the cover and the groove. As the gasket is compressed in the groove, it has groove's dimensions. Gasket dimensions will be noted H_g and L_g.

H = 3,6mm, L=4 mm

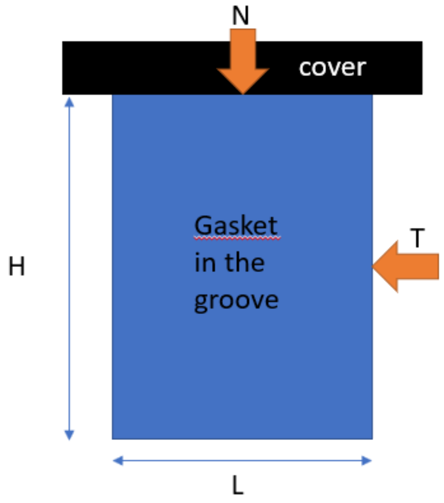


Figure 61: front view (slippage model)

This model is quite rough as it considers that the tangential force T applies on the entire lateral surface of the gasket. Moreover, the groove is not modelized in this model. But this model is a worst case than the reality so it will maximize the chance that the gasket will not slip.

Coulomb's laws:

In order to be sure there is no slipping, we need to verify the equation:

$$T < f \times N$$

Here T comes from the pressure in the circuit and we will say in this model that it applied to the whole gasket lateral surface. (It is not true but as we don't really know where exactly the pression is applied, we are taking less risks by considering it is on the whole surface).

$$T = e \times H \times p_{circuit}$$

N is the contact force between the cover and the gasket. It is related to the compression ratio by Hook's law.

$$N = e \times L \times p_{contact}$$

f is the friction coefficient between elastomer and steel (between 0.25 and 0.45). In order to have the worst case we will take the lower friction coefficient (f = 0.25).

By combining all the equations:

$$p_{contactmin} > \frac{H \times P_{circuit}}{L \times f}$$

For f = 0.45,

$$p_{contactmin} > 1.8 \times P_{circuit}$$

But the worst case is what has to be used for the sizing with f= 0,25:

$$p_{contactmin} > 3{,}6 \times P_{circuit} = 5{,}04\ bar$$

$$p_{contactmin\ after\ ageing} > \frac{p_{contactmin}}{1 - 0.25} = 6{,}72 bar$$

As we took a groove depth of 3,6 mm, we will be able to find a gasket height insuring us that a sufficient contact pressure is applied.

Hook's law: $p_{contact} = E \times \varepsilon$

$$F = p_{contact} \times S$$

S: contact surface btw gasket and cover

$\varepsilon = \frac{\Delta l}{l_0}$ -> $\frac{H - H_g}{H_g}$ with H the final length of the gasket after compression and H_g the initial length of the gasket. Here the final length of the gasket will be the groove depth.

$E = Young\ modulus$

From hook's law we have:

$$H_g = \frac{H}{1 - \frac{p_{contactmini}}{E}} = 4.46\ mm$$

A value of 4.6 mm for the gasket height is fixed to keep a safety margin. Here we have our gasket height (initial height of the gasket). We just need to check if it is in accordance with the compression ratio criteria.

With a groove depth of 3.6 mm and a gasket height of 4.6 mm we have a compression ratio of 0.21. The nominal gasket height and groove depth gives a compression ratio that respect Renault's recommendations (between 0.1 and 0.3). The maximum and minimum compression ratio will be checked later in accordance with tolerances definitions.

Critical analysis of the model:

As said before, this model is not very close from reality but the goal here was to get a value that would maximize the chances that the gasket will not slip in the groove. Some parameters are not considered or could be specified.

For example, the roughness of the cover is not taken into account and it could influence the friction coefficient between the gasket and the groove. As the roughness is not clearly defined and that the link between roughness and friction coefficient is not trivial, the decision to take the worst friction coefficient has been taken.

Moreover, a relaxation coefficient of 0.25 has been fixed but these coefficients are often given by the gasket supplier. Here there are no contacts with gasket supplier, so a value used by Renault for EPDM relaxation has been retained.

Finally, the surface application of the tangential force is the whole lateral surface of the gasket. It may not be true in reality but it is difficult to know so it has been decided to consider the worst case.

4.4.4 Guarantying gasket stability

Another potential source of sealing failure for elastomer gasket is the buckling of the gasket in the groove during the assembly. If the gasket buckle in its groove, the contact between the gasket and the cover may not be maintained and this could open a leak path. Here a minimum gasket width will be defined using a bucking model. Figure 62 shows the model used to characterize the gasket bucking.

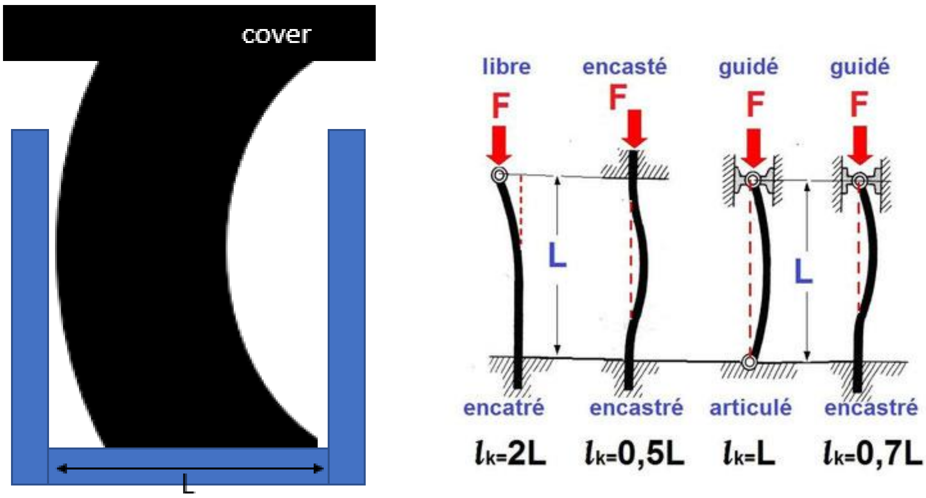


Figure 62: buckling model

The limit force before buckling is below:

$$Fc = \frac{E \times I \times \pi^2}{{l_f}^2}$$

l_f = bucking length

As both extremities of the gasket will stay on the same line but are not embedded lf = Hg with Hg the initial length of the gasket.

I is the moment of inertia of the gasket section. For a rectangle it is $\frac{b \times h^3}{12}$ with b the width and h the height.

With current notations, $I = \frac{e \times L_g{}^3}{12}$, with e a part of the gasket total length. Figure 63 shows what represents e.

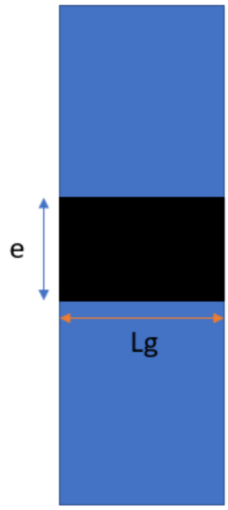


Figure 63: top view of the gasket (buckling model)

Here the goal is to find a gasket width that would ensure that the gasket will not buckle. So, the contact force and the critical buckling force will be compared. The contact force is:

$$F_{contact} = E \times \varepsilon \times e \times L_g$$

If the contact force is higher than the critical force the gasket is more likely to buckle. So, the criteria is $F_{contact} > F_{crit}$. After some manipulations, the limit width of the gasket is:

$$L_g > \sqrt{\frac{12 \times \varepsilon \times L^2}{\pi^2}} = 2.36\ mm$$

L_g final = 2.5 mm to have a confinement ratio as close as possible from 1 and to have a safe margin.

<u>Critical analysis of the model:</u>

This model is not perfect. Normally, this buckling model is used in beam theory. The beam theory is used with some hypothesis. One of them suppose than the one of the beam dimensions is approximately 10 times higher than the two others. This hypothesis is not respected in in our model. Indeed, the gasket length is much higher than gasket width and height. However, this model has been used before by Renault to characterize buckling phenomenon and its results has been compared with trials. The differences between the results from the model and from the trials are quite similar. The critical force on trials are 20% higher than critical force from the model. This means that results from the model can be used and that the model is a worst case than the reality.

4.4.5 Tolerances and comparison with Renault's recommendations:

The gasket dimensions that have been chosen are nominal values. These rough models have been used to give an idea of which dimensions are appropriate for this groove dimensions. Now tolerances will be defined and dimensions with tolerances will be validated or not with Renault's recommendations.

Gasket tolerances and design:

Gasket tolerances has been defined in accordance with the norm ISO 3302-1 [10] for moulded silicon gasket. There will be a tolerance of +/- 0.2 on the gasket height and +/-0.15 on the gasket width. The gasket section will be a rounded rectangle (Figure 64).

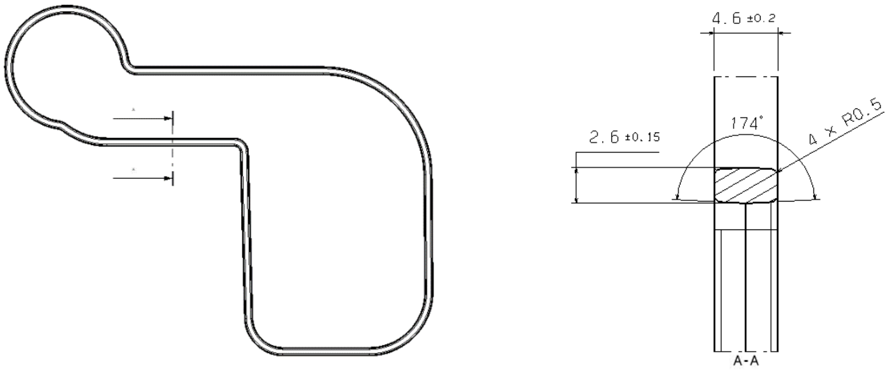


Figure 64: Gasket design

Groove tolerances and design:

The groove will be machined on the housing. According to Renault's experience, tolerances on the groove will be +/- 0.15 mm. Concerning the groove design, the groove is rectangular and there is a chamfer on the edges in order not to endangers operators while assembling the gasket. Chamfer dimensions on the top and on the end of the groove are chosen thanks to Renault's recommendations. Detailed drawing of the groove can be found below with Figure 65.

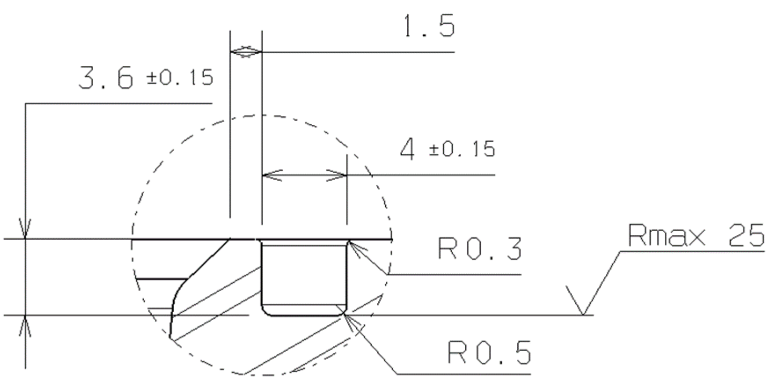


Figure 65: Groove design

Dimensions :

H: 3.6 +/- 0.15 mm
L: 4 +/- 0.15 mm
H_g: 4.6 +/- 0.20 mm
L_g: 2.5 +/- 0.15 mm

Comparison with Renault's criteria:

Covering surface: As fixations holes are behind the groove on the sealing flange, the gasket is automatically always covered, whatever the tolerances.

Compression ratio :
Max: 0.28 < 0.3 = Ok
Min: 0.14 > 0.1 = Ok

Confinement ratio :
Smax gasket: 12.72 mm^2
Smin groove: 13.28 mm^2
Confinement ratio max = 0.96

Every criterion is respected, and this design is fixed for the next part. Drawings of the sealing flange can be found in Annex 3.

4.4.6 Good pressure repartition (simulations to check the geometry)

The objective of this part is to validate a geometry for fixing the cover by trying to stay as close as possible from the initial design. It should be possible as simulations has already been performed on the initial design when trying to complete the failure tree. The only thing that change is the reaction force from the gasket on the cover.

In order to do that, former simulations will be re-used and modified in order to include the pressure applied by the gasket on the cover.

But first, the maximum parting line opening need to be fixed.

4.4.6.1 Maximum parting line opening

In order to find the maximum parting line opening, the Coulomb model will be re-used. Indeed, the maximum parting line opening is reached when the gasket starts slipping on the cover. When the opening between the cover and the carter increase, the normal force decreases, and the tangential force increases. At a certain point, the tangential force will be too strong comparing to the normal force and the gasket will slip on the cover. This point where the gasket starts to slip is the maximum parting line opening. It can be found by looking at the crossing point between the curve f*N(x) and T(x) (Figure 67). (Annex 4)

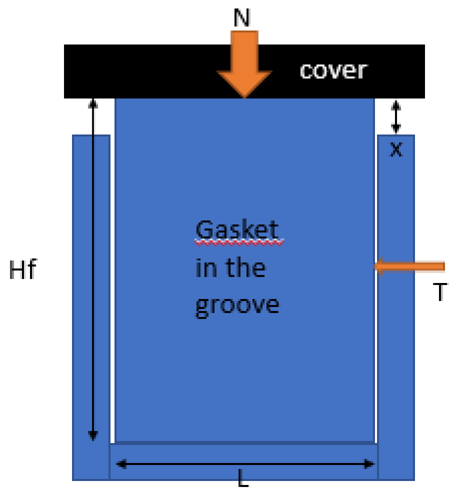


Figure 66: Parting line opening calculations model

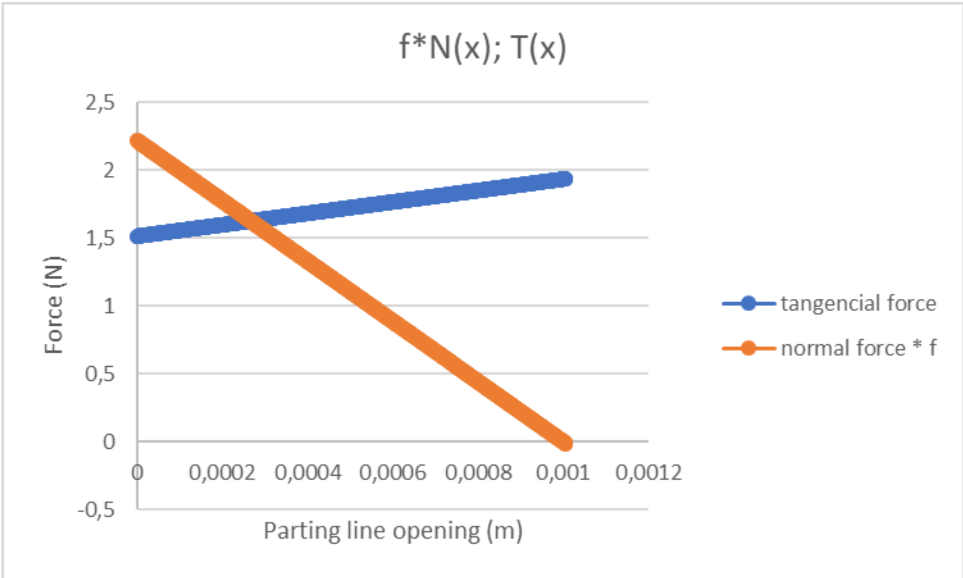


Figure 67: Maximum parting line opening

The two curves cross at a parting line opening of 267 µm. Here the simulations will help us to validate a design. If the results indicate a parting line opening lower than that maximum value, the design will be fixed.

4.4.6.2 Model

The model stays almost the same. The major modification is the addition of the groove on the sealing flange. As the elastomer behaviour is quite difficult to modelized. As the interest is placed in the parting line opening, the gasket will be included in the model by adding a pressure on the cover on the contact surface between the gasket and the cover.

- Addition of the groove on the 3D model (Figure 68)

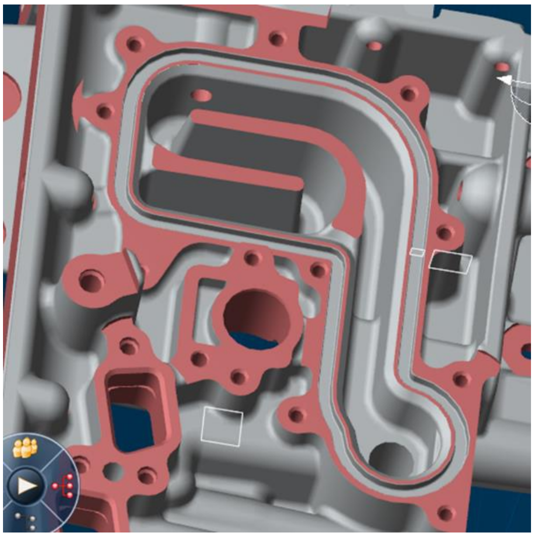


Figure 68: Housing 3D model with the groove

- Addition of a pressure (gasket pressure)

Figure 69: pressure modelisation (elastomer model)

The circuit pressure is applied on the blue surface on the left and the gasket pressure is applied on the blue surface on the right (Figure 69).

To conclude, this model supposes that the pressure from the gasket on the cover is constant. In reality it is not true because as the cover opens, the compression ratio decreases, and the pressure decreases too. This model is a worst case than the reality and will give us a parting line opening higher than the reality.

4.4.6.3 Results

- Parting line opening

Result is an opening of 41 µm. However, this opening is not on the sealing flange. The maximum opening on the sealing flange is approximately 37 µm (red circle on the Figure 70). This value is much lower than the limit of 267 µm. This fixations design, which is the same as the former one, can be kept.

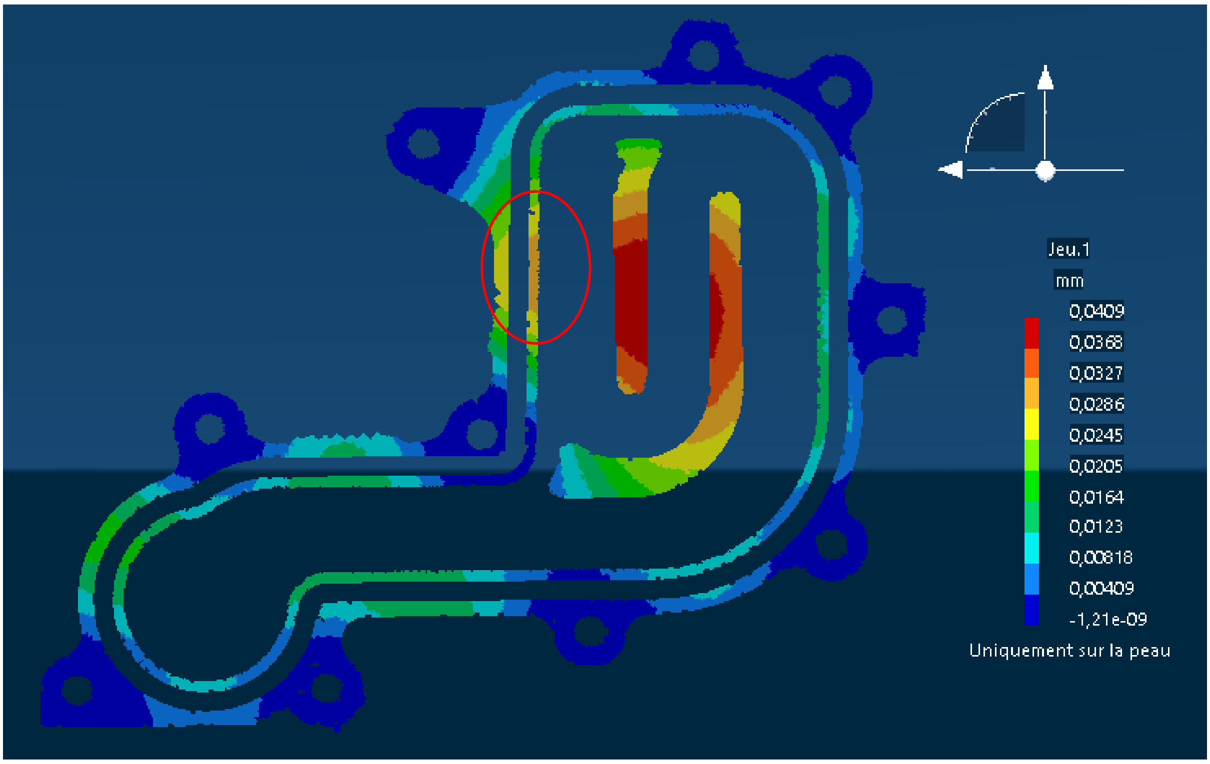


Figure 70: Parting line opening (elastomer model)

4.4.7 Cost comparison

What could be interesting is to try to estimate the cost difference between FIPG technology and elastomer gasket in a groove technology for this application. This will be an estimation as there is no contacts with suppliers for the gasket production.

The major difference between the two sealing technologies is that elastomer gasket in a groove needs a mould that is often worth several thousand of euros. On the other hand, FIPG needs a robot to apply the silicon on the sealing flange. However, Renault already have these robots.

Taking into consideration all these parameters, the conclusion is that for 10000 produced sealings, elastomer gasket in a groove technology is about 20 times more expensive than FIPG technology.

This cost difference can explain the initial choice of FIPG technology. Elastomer gasket in a groove is preferred when there is a real need of disassembly after sale.

4.4.8 Conclusion

As the design respects every criterion, it can be fixed. This design should ensure us of an efficient sealing under operating conditions (no slipping of the gasket and good gasket stability). Models used to size the gasket and the groove have always been worst cases than the reality and that is why the design may be a little oversized.

However, precise values of relaxation coefficient, friction coefficient or young modulus could not be used as elastomers are very complicated materials. Results would have been more precise in case of a collaboration with elastomer gasket suppliers. Indeed, they can model their elastomer gasket in a groove and get more precise results.

Finally, some improvements could be considered on the design. On the benchmarking part, assembly feasibility problems have been mentioned. It shows that pips on the gasket can be used to increase the stiffness of the gasket and make the assembly easier. A positioning tab could also be used to facilitates the assembly of the gasket in its groove. Those design particularities are quite hard to model and validate without trials. Moreover, there is no need for chamfer on the sealing flange. Those could be removed but as it is a proposition of modification, chamfer on the housing cannot be removed.

Conclusion

Technic benefits

This internship gave me the opportunity to learn the technical knowledge related to engine sealing technologies. As a first step, the goal was to get this knowledge from technical support concerning sealing types. These documents help designer to conceive sealing depending on the application, the solicitations, and the economic constraints. This was a way to discover the three important sealing principles. The benchmarking was a way to review all the engine architecture and more precisely sealing types by applications. Moreover, it permitted me to get more into sizing parameters for elastomer gasket and FIPG.

After getting this knowledge, I started the study case. The first part was the investigation of some possible root causes. During this part I had the opportunity to learn to use the simulation module from Catia V6. I learned to model sealing assembly for static simulations for two sealing types. This has been a way to get to know more about Catia V6 as it is a bit different from Catia V5.

Finally, the final part on the re-sizing of the sealing with elastomer gasket in a groove permitted me to do a pre-sizing from the beginning (functional demand) to the end (first drawings and 3D models). The all approach was very interesting as I had to make choices on my own and be able to justify them. The exercise would have been easier and more precise if I had been able to work with suppliers. Indeed, for moulded gasket with complex geometry, there are no norms on the section or dimensions as it can be found for O-ring for example.

Personal inputs

From a personal point of view, the Covid-19 context made it difficult to assimilate completely in the team as I spent 5 months working from home. As a trainee when we ask for information, we are not a priority and it was even more true in this context. That is why I learned to be persuasive and persistent to get as much information as I needed.

Working from home forced me to get more autonomous as it was more difficult to contact or ask questions directly.

Obviously, I have some regrets concerning this internship as I would have wanted to get to know more my co-workers but also because it was planned to do a bit more including some trials on sealing for E-PWT. However, this has been a great experience, I learned a lot from it and will certainly use what I learnt for my next experiences.

Annex 1

Here is an example of measurement on excel. The sealing flange geometry parameters are measured with coloured arrows. Green is for the sealing flange length, red for the overhang length, blue for the chamfer width and black for the span length. Measurements are done between each fixation holes

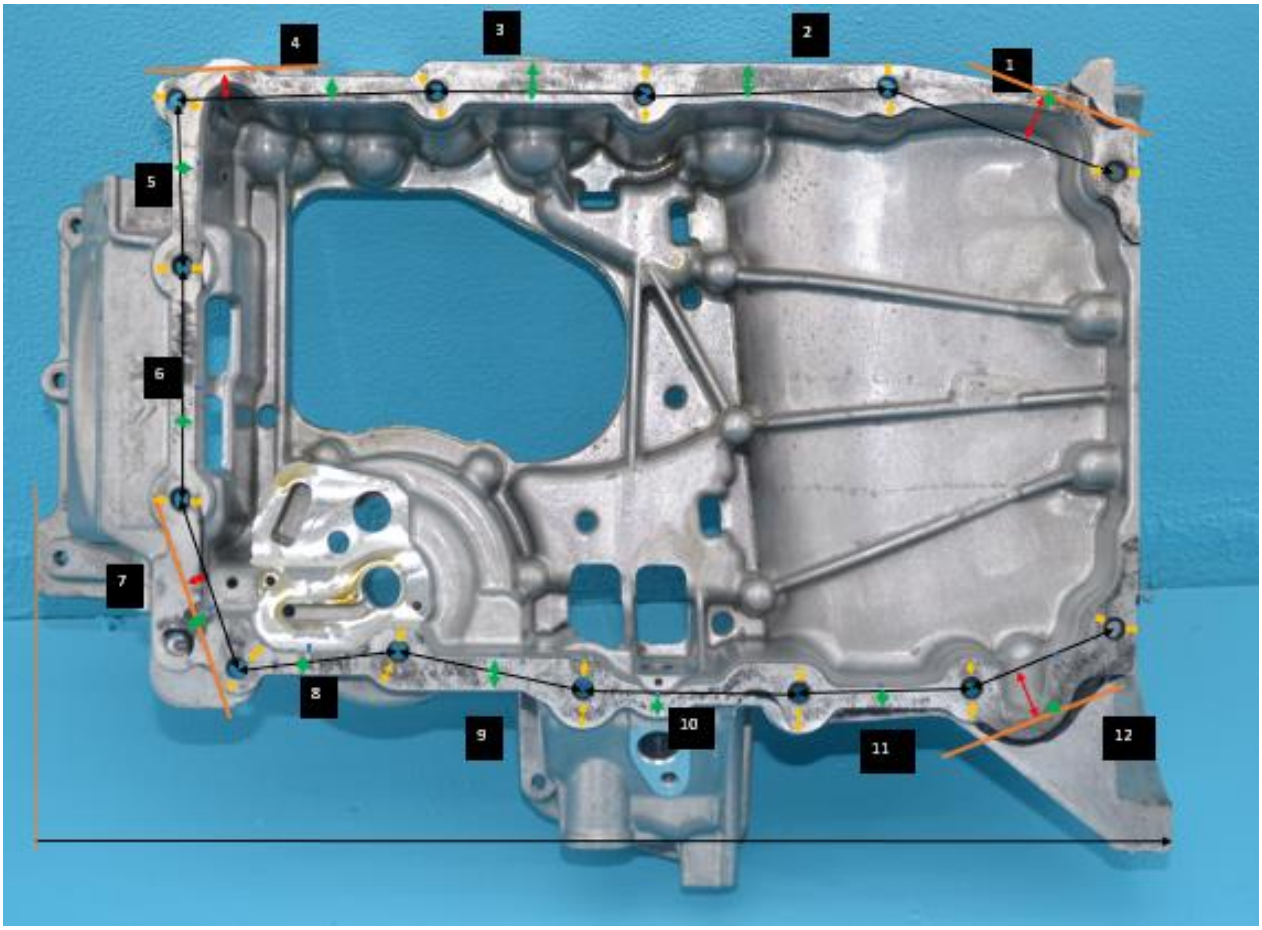


Figure 71: Benchmarking measurement example

The long black arrow on the bottom of the part is the reference to set the scale distance.

After placing arrows, their height and width can be found.

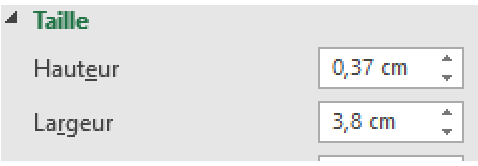


Figure 72: Arrows dimensions on excel (benchmarking)

Then, by a simple Pythagorean theorem, total length of the arrow can be found.

Annex 2

Failure	Causes	Sub-causes		Effects	Checking tools	Checking status	Values
Water leakage from FIPG surface (cold sealing) --> parting line opening	Geometry	sealing flange geometry	chamfer length too short	insufficient elastic reserve / particle detachment	measurement on 3D model/ comparison with Renault's recommendations	✓	
			sealing flange length too short	weaker barrier for the fluid		✓	
			span bolt too long	bad clamping force repartition		✓	
			overhang too long			✓	
			covering length tolerances	covering length too short	Tolerance stack up	☐	
		cover geometry	cover width too low	high deformations	Simulations on Catia	✓	
		bad roughness definition of the sealing flange	R5-25, Rx30	adhesive efficiency	roughness definition	✗	Rmax = 25
		flatness			Tolerance stack up	☐	
	process/ assembly	curing time before tests			Process analysis	✗	20 min btw assembly and pressure
		Surface pollution	detergent residue	adhesion problem		☐	
		scratch on the sealing flange				✓	
		bad clamping	clamping order	cover deformations		✓	
			clamping torque	clamping force repartition		✓	
		roughness of the surface not respecting definition		adhesive problem	Roughness measurement	✗	R0,8, Rx 4,5
		delivery path	bad positioning of the FIPG cord	covering length too short	Process analysis	✓	
		operator mishandling	Difficulties during the assembly	sealing deterioration		✗	operator

Figure 73: Full failure tree (PEC)

This tree summarizes every investigation on the PEC leak. The checking status column indicates if the possible root cause has been investigated, and if yes it confirms if it respects recommendations. The column value indicates what is the problem if a root cause has been identified.

Annex 3

Sealing flange dimensions are not detailed. As it can be seen on the groove drawing, the groove

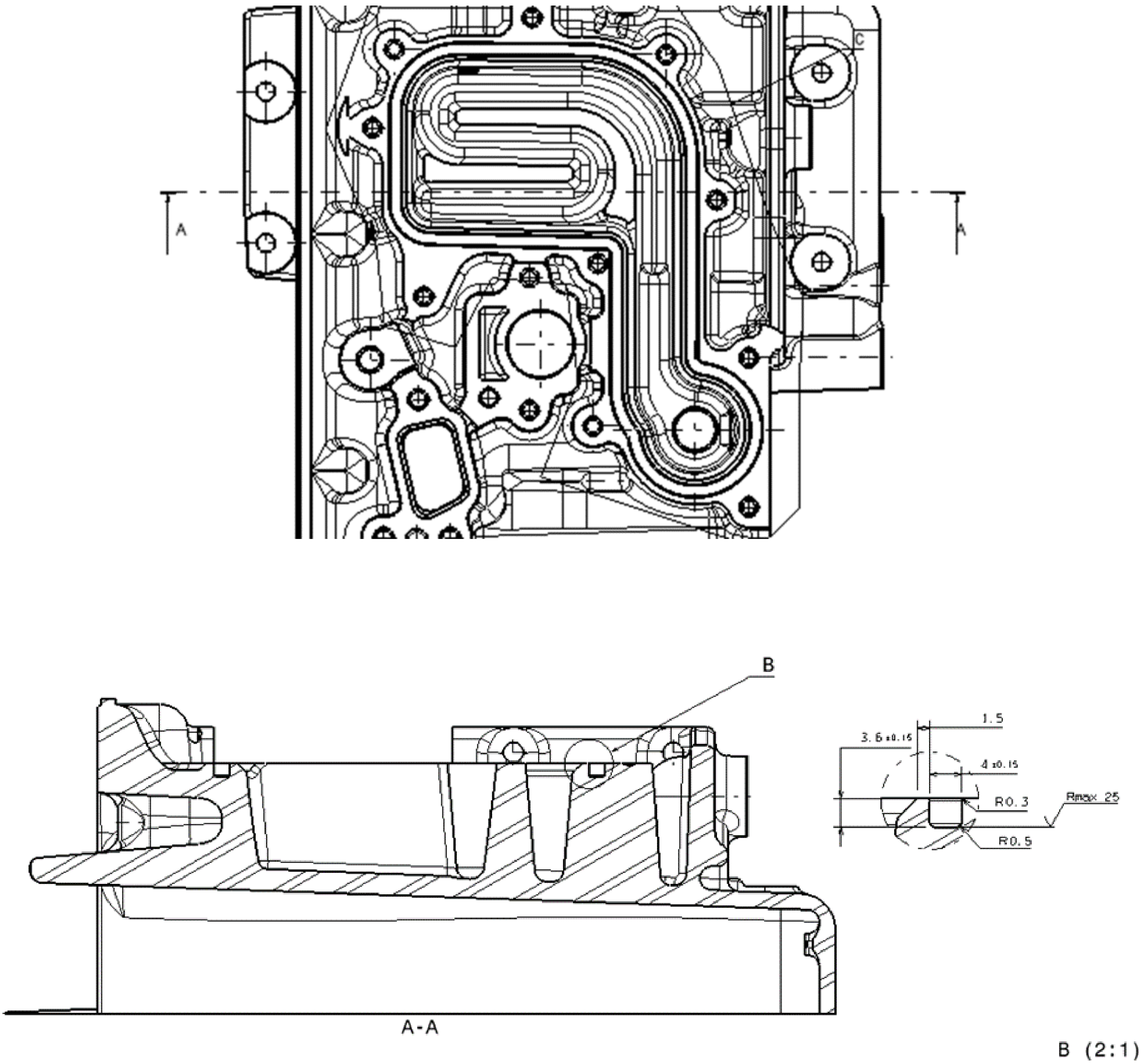
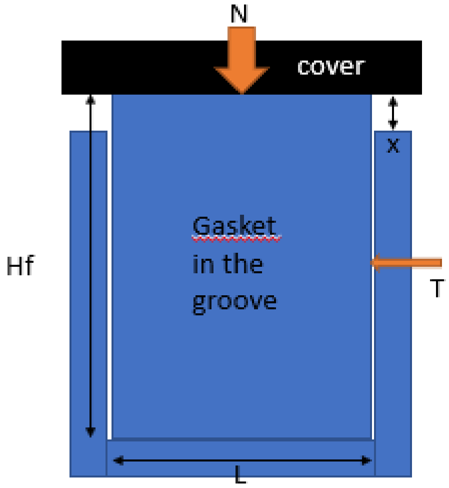


Figure 74: Sealing flange drawings (new PEC)

is positioned in regard to the sealing flange edge. There is 1.5 mm between the groove and the sealing flange edge. This value has been chosen to be sure that the groove will fit on the sealing flange. It is important to check that there are no collisions between the groove and fixations holes. By considering positioning tolerances (fixations holes) and groove width tolerances, the groove position on the sealing has been validated.

Annex 4



The detailed calculations of N(x) and T(x) can be found below. N depends on x value because the compression ratio depends on the parting line opening x.

$$N(x) = p_{contact}(x) \times S$$

$$N(x) = E \times \varepsilon(x) \times S \text{ with } \varepsilon(x) = \frac{(H_i-(x+H_f))}{H_i}$$

$$\mathrm{N(x)} = \mathrm{e} \times \mathrm{L} \times \mathrm{E} \times \frac{(-x - H_f + H_i)}{H_i}$$

T depends on x value because the surface on which T the pressure is applied increase with the parting line opening x.

$$T(x) = S(x) \times p_{circuit}$$

$$\mathrm{T(x)} = (H_f + x) \times e \times P_{circuit}$$

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Illustration Table